TRANSACTIONS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 59

Fifty-Ninth Annual Meeting
Chicago, Ill.
January 26-29, 1953

Semi-Annual Meeting
Denver, Colo.
June 29-30, July 1, 1953

PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE EXECUTIVE SECRETARY
62 Worth St., New York 13, N. Y.
Printed in U.S.A.

Engin. Library
TH
7201
.A52
v.59

Copyright 1954
by
THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Printed and bound by
The Horn-Shafer Co.
Baltimore, Md.
U.S.A.

Engin. Lib
Factory
11.10.54

CONTENTS

CHAPTER		PAGE
1462	**Fifty-Ninth Annual Meeting, 1953, Chicago, Illinois**	1
1463	A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment, by Tohru Inouye, F. K. Hick, M.D., R. W. Keeton, M.D., J. Losch and Nathaniel Glickman	35
1464	Two Methods of Calculating the Friction in Pipe Lines, by F. E. Giesecke	49
1465	Compression Tank Selection for Hot Water Heating Systems, by H. A. Lockhart and G. F. Carlson	55
1466	The Bacteria and Odor Control Problem in Occupied Spaces, by R. L. Kuehner	77
1467	Heat Transmitted to the I=B=R Research Home from the Inside Chimney, by W. S. Harris and R. J. Martin	97
1468	Fuel Consumption Analysis for Multi-Family Housing Projects, by Rutcher Skagerberg and J. E. Phifer	113
1469	Friction Losses in Round Aluminum Ducts, by F. W. Hutchinson	127
1470	Pressure Losses in 4-inch Diameter Galvanized Metal Duct and Fittings, by H. G. Conn, W. G. Colborne and W. G. Brown	139
1471	Room Air Distribution Research for Year 'Round Air Conditioning—Part I—Supply Outlets at One High Sidewall Location, by S. F. Gilman, H. E. Straub, A. E. Hershey and R. B. Engdahl	151
1472	Field Studies of Floor Panel Control Systems, by A. B. Algren, E. F. Snyder, Jr., and J. S. Locke	173
1473	Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee and C. M. Humphreys	197
1474	The Shading of Sunlit Glass—An Experimental Study of Slat-Type Sun Shades, by G. V. Parmelee, W. W. Aubele and D. J. Vild	221
1475	Maximum Downward Travel of Heated Jets from Standard Long Radius *A.S.M.E.* Nozzles, by Linn Helander, Shee-Mang Yen and R. E. Crank	241
1476	Air Velocities in Ventilating Jets, by G. L. Tuve	261
1477	Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit, by H. T. Gilkey, D. R. Bahnfleth and R. W. Roose	283

CHAPTER		PAGE
1478	**Semi-Annual Meeting, 1953, Denver, Colorado**	305
1479	Performance of a Forced-Draft Cooling Tower, by B. H. Spurlock, Jr.	311
1480	Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F, by Tohru Inouye, F. K. Hick, M.D., S. E. Telser, M.D., and R. W. Keeton, M.D.	329
1481	Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems, by W. L. Rogers	347
1482	Electrical Analogger Application to the Heat Pump Process, by Carl F. Kayan	361
1483	Design and Performance of a Residential Earth Heat Pump, by Merl Baker	371
1484	Moisture Movement in Soils Due to Temperature Difference, by W. A. Hadley and Raymond Eisenstadt	395
1485	Design Data for Slat-Type Sun Shades for Use in Load Estimating, by G. V. Parmelee and D. J. Vild	403
1486	Automatic Permeance Measurement by the Permeometer, by F. A. Joy and A. W. Sherdon	435
1487	Thermal Performance of Frame Walls—Part II—Air Spaces Blocked at Mid-Height, by G. O. Handegord and N. B. Hutcheon	449
1488	Development of the Thermal Conductivity Probe, by F. C. Hooper and S. C. Chang	463
1489	Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence, by H. T. Gilkey, R. W. Roose and M. E. Childs	473
1490	Heat Exchanges in a Floor Panel Heated Room, by L. F. Schutrum, G. V. Parmelee and C. M. Humphreys	495
1491	Further Studies of the Thermal Characteristics of Plaster Panels, by L. F. Schutrum and C. M. Humphreys	511
1492	Heat Flow Analysis in Panel Heating or Cooling Sections—Case II—Floor Slab on Earth with Uniformly Spaced Pipes or Tubes at the Slab-Earth Interface, by H. B. Nottage, C. V. Franks, L. E. Hulbert and L. F. Schutrum	527
	In Memoriam	549
	Homer Addams (1873-1953)	551
	Dr. Frederick E. Giesecke (1869-1953)	553
	Louis A. Harding (1876-1953)	555
	Gustus L. Larson (1881-1953)	557
	Index	559

Officers and Council—1953

The American Society of Heating and Ventilating Engineers

Headquarters: 62 Worth St., New York 13, N. Y.
(Tel.: WHitehall 3-0377)

OFFICERS

President.....Reg. F. Taylor
First Vice President.....L. N. Hunter
Second Vice President.....J. E. Haines
Treasurer.....John W. James
Executive Secretary.....A. V. Hutchinson
Technical Secretary.....Carl H. Flink
Assistant Secretary.....David T. Donovan

COUNCIL

Reg. F. Taylor, *Chairman* — L. N. Hunter, *Vice Chairman*

Three Years	*Two Years*	*One Year*
I. W. Cotton	M. K. Fahnestock	B. L. Evans
A. W. Edwards	P. B. Gordon	J. H. Fox
L. T. Mart	R. T. Kern	N. H. Peterson
E. R. Queer	D. M. Mills	B. H. Spurlock, Jr.
		Ernest Szekely

ADVISORY BOARD

Ernest Szekely, *Chairman;* Lester T. Avery, M. F. Blankin, S. E. Dibble, S. H. Downs, E. O. Eastwood, W. L. Fleisher, H. P. Gant, F. E. Giesecke, E. Holt Gurney, L. A. Harding, H. M. Hart, C. V. Haynes, D. D. Kimball, G. L. Larson, S. R. Lewis, A. J. Offner, F. B. Rowley, L. E. Seeley, A. E. Stacey, Jr., G. L. Tuve, A. C. Willard, C.-E. A. Winslow, and B. M. Woods.

COUNCIL COMMITTEES

Executive: Ernest Szekely, *Chairman;* L. N. Hunter, John W. James.
Sub-Committee: H. E. Sproull, *Chairman;* C. S. Koehler, E. L. Crosby.

Finance: N. H. Peterson, *Chairman;* J. E. Haines, E. R. Queer, John W. James, *Ex-Officio.*
Ways and Means: E. R. Queer, *Chairman;* L. T. Mart, M. R. Eastin, M. K. Fahnestock, I. W. Cotton.
Investment Policy: E. N. McDonnell, *Chairman;* M. F. Blankin, A. J. Offner.

Membership: D. M. Mills, *Chairman;* R. T. Kern, A. W. Edwards.

Program and Papers: B. H. Spurlock, Jr., *Chairman;* P. B. Gordon, M. K. Fahnestock.

Standards: L. N. Hunter, *Chairman;* P. B. Gordon, B. H. Spurlock, Jr.

GENERAL COMMITTEES

Admission and Advancement: Roswell Farnham, *Chairman (one year);* Carl F. Boester *(two years);* W. A. Kuechenberg *(three years).*

Publication: P. J. Marschall, *Chairman (one year);* Linn Helander *(two years);* Axel Marin *(three years).*

Guide: W. S. Harris, *Chairman;* F. J. Nunlist, Jr., C. D. Shields *(one year);* P. R. Achenbach, R. E. Cherne, A. B. Newton *(two years);* W. M. Wallace, II, A. L. Hesselschwerdt, D. L. Mills *(three years);* R. S. Dill, *Ex-Officio.*

Charter and By-Laws: G. W. F. Myers, *Chairman (one year);* G. D. Winans *(two years);* M. W. Bishop *(three years).*

F. Paul Anderson: L. N. Hunter, *Chairman;* M. F. Blankin, J. C. Fitts, L. G. Miller, L. P. Saunders.

Chapter Relations: B. L. Evans, *Chairman;* E. L. Crosby, E. T. Gessell, D. M. Hummel, W. B. Morrison, C. H. Pesterfield, William Philip.

Chapters Conference: G. W. Bornquist, *Chairman;* W. J. Collins, Jr., *Vice Chairman;* J. S. Burke, *Secretary.* Members and Alternate Members selected by local chapters are respectively:

Chapter	*Member*	*Alternate Member*
Arkansas	R. E. Blaylock	J. L. Brown
Atlanta	J. G. Croley	J. M. Lazenby
Baltimore	E. R. Kent	H. D. Glaser
British Columbia	M. E. Minaker	C. W. Leek
Central New York	C. R. Acheson	K. T. Sprague
Central Ohio	A. D. Bogen	N. T. Hess
Cincinnati	H. E. Russell	R. C. Beineke
Connecticut	A. J. Lawless	T. L. Arnold
Delta	J. S. Burke	Walter Cooke
Empire State Capital	H. Horowitz	N. W. Burrill
Golden Gate	T. J. White	J. E. Murray
Illinois	G. W. Bornquist	H. G. Chapin
Indiana	F. C. Barton	J. T. Hardin
Inland Empire	G. M. Dieter	J. A. Doyle
Iowa	R. S. Stover	R. H. Schnell
Kansas	R. L. Pennington	R. F. Bauer
Kansas City	G. H. Stoffer	F. K. Ladewig
Manitoba	A. K. Piercy	G. E. Randall
Massachusetts	G. B. Torrens	G. D. Fife
Memphis	M. H. Ellis	J. W. McIntosh
Miami Valley	D. L. Bergman	R. J. Perkins
Michigan	C. A. Strand	R. H. Oberschulte
Minnesota	William Sturm	V. E. Pearson
Montreal	W. G. Hole	B. J. Horsburgh
Nebraska	O. J. Smith	H. H. Ulrich
New York	P. B. Gordon	J. E. Schechter
Northeastern Oklahoma	C. H. Dollmeyer	J. N. Watt
Northern Ohio	J. M. Black	L. C. Burkes
Northern Piedmont	J. E. Hart	S. T. Oliver
North Jersey	F. H. Faust	C. H. Smith
North Texas	H. G. Gregerson	C. R. Gardner

GENERAL COMMITTEES—1953 *(Continued)*

Chapter	*Member*	*Alternate Member*
Oklahoma City	W. J. Collins, Jr.	F. R. Denham
Ontario	N. W. Kingsland	J. H. Ross
Oregon	A. N. Hoss	B. W. Farnes
Ottawa Valley	G. L. Ostiguy	W. J. Robinson
Pacific Northwest	H. T. Orebaugh	T. C. Caskey
Philadelphia	M. E. Barnard	E. K. Wagner
Pittsburgh	C. H. Schneider	E. H. Riesmeyer, Jr.
Rocky Mountain	Fred Janssen	B. H. Spurlock, Jr.
Sacramento Valley	L. A. O'Meara	E. C. McKinsey
St. Louis	C. H. Burnap	H. C. Sharp
Shreveport	M. A. Hudson	W. E. Fitzgerald
South Texas	J. W. Holland	W. N. Pauley
Southern California	J. S. Earhart	L. B. Davenport
Southern Piedmont	R. E. Mason	R. O. McGary
Southwest Texas	L. H. Hornor, Jr.	D. E. Locher
Utah	M. L. Gollaher	Alfred Richeda
Virginia	E. D. Duval	H. E. Marquart
Washington, D. C.	G. R. Walz	J. G. Muirheid
Western Michigan	R. W. Mangrum	H. R. Limbacher
Western New York	J. M. Quackenbush	C. W. Stone
Wisconsin	F. J. Nunlist, Jr.	H. F. Brinen

Committee on Research—1953

A.S.H.V.E. Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio
(Tel.: EXpress 6161)

R. S. Dill, *Chairman*

B. H. Jennings, *Vice Chairman* C. M. Humphreys, *Acting Director of Research*

Three Years	*Two Years*	*One Year*
R. S. Dill	I. W. Cotton	A. B. Algren
F. H. Faust	W. A. Grant	John Everetts, Jr.
R. W. Keeton, M.D.	N. B. Hutcheon	A. J. Hess
H. A. Lockhart	B. H. Jennings	T. H. Smoot
J. W. McElgin	C. O. Mackey	W. N. Witheridge

Executive Committee: R. S. Dill, *Chairman;* B. H. Jennings, *Vice Chairman;* John Everetts, Jr., F. H. Faust, C. O. Mackey.

Long-Range Research Program: B. H. Jennings, *Chairman;* John Everetts, Jr., F. H. Faust, John W. James, C. O. Mackey, R. S. Dill, *Ex-Officio.*

Technical Advisory Committees—1953

Air Cleaning: A. B. Algren*, *Chairman;* R. B. Crepps, L. L. Dollinger, Jr., O. C. Eliason, B. G. Evans, R. S. Farr, C. D. Graham, A. J. Hess*, A. B. Hubbard, G. F. Landgraf, K. W. MacKenzie, Arthur Nutting, H. E. Robinson, S. L. Root, Jr., C. B. Rowe, W. T. Van Orman, R. P. Warren, W. B. Watterson.

* Member of Committee on Research.

GENERAL COMMITTEES—1953 *(Continued)*

Air Distribution: G. B. Priester, *Chairman;* W. A. Grant*, *Vice Chairman;* J. M. Black, H. F. Brinen, R. M. Conner, S. H. Downs, Nathan Feder, S. F. Gilman, Linn Helander, F. B. Holgate, W. O. Huebner, W. W. Kennedy, E. J. Kurek, R. D. Madison, G. E. McElroy, L. G. Miller, H. B. Nottage, C. H. Randolph, G. L. Tuve.

Combustion: T. H. Smoot*, *Chairman;* P. R. Achenbach, R. B. Engdahl, H. G. Hays, R. C. Johnson, W. B. Kirk, H. R. Limbacher, D. W. Locklin, A. A. Marks, W. M. Myler, Jr., E. A. Norman, Jr., E. O. Olson, K. O. Schlentner, R. K. Thulman, E. C. Webb, D. L. Wooden.

Cooling Load: C. O. Mackey*, *Chairman;* K. O. Alexander, John Everetts, Jr.*, L. F. Fisher, W. F. Friend, H. T. Gilkey, R. H. Heilman, H. W. Heisterkamp, A. J. Hess, R. C. Jordan, A. T. Jorn, C. F. Kayan, J. N. Livermore, P. E. McNall, Jr., I. A. Naman, H. B. Nottage, C. P. Roberts, J. P. Stewart, H. B. Vincent, W. E. Zieber.

Evaporative Cooling: A. J. Hess*, *Chairman;* R. S. Ash, H. E. Degler, S. F. Duncan, H. S. Dutcher, John Engalitcheff, Jr., Stuart Giles, E. V. Gritton, Leo Hungerford, R. C. Jordan, L. C. Pellegrini, R. J. Salinger, Edward Simons, W. T. Smith, B. H. Spurlock, Jr., D. D. Wile.

Heat Flow Through Glass: R. A. Miller, *Chairman;* N. B. Hutcheon*, *Vice Chairman;* W. J. Arner, F. L. Bishop, E. W. Conover, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, D. R. Muir, H. B. Vincent.

Heat Pump: E. P. Palmatier, *Chairman;* F. H. Faust*, E. R. Ambrose, F. R. Ellenberger, W. F. Friend, S. F. Graziano, G. C. Hall, R. C. Jordan, C. F. Kayan, J. D. Kroeker, F. R. O'Brien, J. R. Swanton, Jr.

Heating Load: P. R. Achenbach, *Chairman;* H. A. Lockhart*, A. I. Brown, W. F. Friend, W. S. Harris, J. M. Hartman, K. L. Lamm, L. G. Miller, V. L. Miller, T. F. Rockwell, S. I. Rottmayer, C. W. Signor.

Hot Water and Steam Heating: John W. James, *Chairman;* H. A. Lockhart*, *Vice Chairman;* R. C. Chewning, H. C. Day, W. S. Harris, L. N. Hunter, A. T. Jones, E. G. Keller, M. W. McRae, N. D. Skinner, S. K. Smith, Benjamin Spieth, M. H. Westerberg.

Human Calorimetry: G. L. Tuve, *Chairman;* I. W. Cotton*, *Vice Chairman;* Nathaniel Glickman, J. D. Hardy, E. R. Queer, L. E. Seeley, C. P. Yaglou.

Industrial Environment: P. J. Marschall, *Chairman;* J. W. McElgin*, *Vice Chairman;* J. L. Alden, A. D. Brandt, F. H. Buzzard, F. N. Calhoon, K. J. Caplan, J. H. Clarke, R. B. Foley, T. F. Hatch, W. C. L. Hemeon, W. O. Huebner, J. M. Kane, M. G. Kershaw, G. E. Klapper, G. E. McElroy, H. A. Mosher, K. E. Robinson, Leslie Silverman, B. R. Small, R. P. Warren, W. N. Witheridge*, H. E. Ziel.

Insulation: M. W. Keyes, *Chairman;* N. B. Hutcheon*, *Vice Chairman;* L. A. Barron, C. B. Bradley, P. D. Close, R. B. Crepps, E. T. Erickson, G. A. Erickson, R. H. Heilman, F. A. Joy, W. A. Knudson, H. E. Lewis, C. E. Lund, W. J. Marshall, V. L. Miller, F. E. Parsons, E. L. Perrine, K. M. Ritchie, H. E. Robinson, A. S. Russell, T. D. Stafford, L. V. Teesdale, R. K. Thulman.

Odors: T. H. Urdahl, *Chairman;* W. A. Grant*, *Vice Chairman;* L. H. Beck, N. E. Berry, V. G. Dethier, A. H. Gee, A. B. Hubbard, G. W. Meek, Sherman Ross, R. G. H. Siu, Amos Turk, W. N. Witheridge*.

---

* Member of Committee on Research.

GENERAL COMMITTEES—1953 *(Concluded)*

Panel Heating and Cooling: P. B. Gordon, *Chairman;* C. O. Mackey*, *Vice Chairman;* A. B. Algren*, John Everetts, Jr.*, L. N. Hunter, R. L. Maher, S. K. Smith, E. F. Snyder, Jr., J. M. van Nieukerken.

Group A—Heat and Distribution Within and Behind the Panel: R. L. Maher, *Chairman;* H. L. Flodin, G. D. Lain, V. L. Miller, D. L. Mills, C. W. Nessell, S. I. Rottmayer, R. T. Schoerner, S. K. Smith, O. H. Storey, Jr., R. K. Thulman.

Group B—Heat Transfer Between the Panel and the Space: J. M. van Nieukerken, *Chairman;* C. O. Mackey*, *Vice Chairman;* C. M. Ashley, W. P. Chapman, W. S. Harris, Linn Helander, E. S. Howarth, A. T. Jorn, C. F. Kayan, H. A. Lockhart*, W. E. Long, R. L. Maher, R. A. Miller, D. L. Mills, G. L. Wiggs.

Group D—Controls: E. F. Snyder, Jr., *Chairman;* A. B. Algren*, *Vice Chairman;* H. W. Alyea, Louis Barfus, S. D. Browne, O. P. Bullock, C. A. Gustafson, W. J. Hajek, W. H. Kliever, H. T. Kucera, J. S. Locke, J. F. McCauley, P. F. Neess, J. F. Page, L. C. Plaehn, J. K. M. Pryke, A. O. Roche, Jr., C. W. Signor.

Physiological Research: M. K. Fahnestock, *Chairman;* R. W. Keeton*, M.D., *Vice Chairman;* Lester T. Avery, Thomas Bedford, H. S. Belding, William Bruce, A. C. Burton, A. P. Gagge, Nathaniel Glickman, F. K. Hick, M.D., John W. James, R. E. Johnson, R. D. Madison, P. J. Marschall, Sid Robinson, C. L. Taylor, T. L. Willmon, C.-E. A. Winslow.

Plant and Animal Husbandry: A. J. Hess*, *Chairman;* A. B. Algren*, H. A. Borthwick, Samuel Brody, John Everetts, Jr.*, F. G. Gustafson, S. R. Lewis, H. A. Lockhart*, H. H. Mitchell, K. V. Thimann, W. G. Whaley, R. B. Withrow, H. E. Ziel.

Sensations of Comfort: C. S. Leopold, *Chairman;* John Everetts, Jr.*, *Vice Chairman;* Lester T. Avery, M. K. Fahnestock, Nathaniel Glickman, A. J. Hess*, A. C. Menke.

Sorption: G. L. Simpson, *Chairman;* John Everetts, Jr.*, *Vice Chairman;* P. R. Achenbach, G. C. F. Asker, O. D. Colvin, F. C. Dehler, W. E. Emley, Jr., A. S. Gates, Jr., E. W. Gifford, E. R. McLaughlin.

Sound and Vibration Control: H. A. Lockhart*, *Chairman;* C. M. Ashley, A. E. DeSomma, Sidney Gordon, W. F. Hagen, H. C. Hardy, E. M. Herrmann, F. B. Holgate, A. F. Hubbard, G. C. Kerr, R. D. Madison, W. L. Rogers, T. A. Walters.

Weather Data: John Everetts, Jr.*, *Chairman;* J. L. Blackshaw, L. L. Hamig, S. A. Heider, W. L. Holladay, C. O. Mackey*, H. C. S. Thom, M. K. Thomas.

•

Public Relations: J. R. Vernon, C. A. McKeeman, G. H. LaRoi, II, J. H. Fox.

SPECIAL COMMITTEES—1953

A.S.H.V.E. Committee on Code for Testing and Rating Heavy Duty Furnaces: E. K. Campbell, *Chairman;* Bowen Campbell, R. S. Dill, A. P. Loeb, C. J. Low, W. J. MaGirl, B. F. McLouth, F. L. Meyer, L. G. Miller, A. A. Olson, H. A. Pietsch, O. J. Ress, H. A. Soper.

* Member of Committee on Research.

SPECIAL COMMITTEES—1953 *(Concluded)*

A.S.H.V.E. Committee on Safety Regulations for Heating, Ventilating and Air Conditioning Systems: G. P. Nachman, *Chairman;* F. H. Buzzard, H. W. McKenzie, B. F. McLouth, C. H. Randolph.

Awards: Linn Helander, *Chairman;* H. E. Sproull, A. E. Stacey, Jr.

Building: A. J. Offner, *Chairman;* Albert Buenger, P. B. Gordon, L. E. Seeley, T. H. Urdahl.

Engineers' Joint Council: L. E. Seeley (A.S.H.V.E. Representative).

Exposition Rates: E. N. McDonnell, *Chairman;* S. H. Downs, L. N. Hunter.

International Joint Committee on Psychrometric Data: C. O. Mackey, Ithaca, N. Y., *Chairman;* B. A. Dmitrieff, New York, N. Y., *Secretary;* H. H. Bruce, London; G. A. Bull, London; C. S. Cragoe, Washington, D. C.; John A. Goff, Philadelphia; L. P. Harrison, Washington, D. C.; T. J. G. Henry, Toronto; B. H. Jennings, Evanston, Ill.; F. G. Keyes, Cambridge, Mass.; R. F. Legget, Ottawa; S. G. Rison, Washington, D. C.; P. A. Sheppard, London; L. L. York, Ann Arbor, Mich.

Joint Committee for Revision of Code of Minimum Requirements for Comfort Air Conditioning: W. L. Fleisher, *Chairman;* (A.S.H.V.E.) O. W. Armspach, A. C. Buensod, C. C. Cheyney; *(A.S.M.E.)* C. F. Kayan; *(A.S.R.E.)* F. H. Faust, C. F. Holske, J. A. Schurman; *(A.R.I.)* M. M. Lawler, R. A. Gonzalez; Alternate: J. R. Schreiner; (U. S. Public Health Service) W. C. Cooper, M.D.; At Large: M. K. Fahnestock, P. J. Marschall, G. E. May, A. B. Newton; (Medical Profession) R. W. Keeton, M.D., W. J. McConnell, M.D.

Long-Range Planning Committee: J. E. Haines, *Chairman;* P. B. Gordon, A. J. Hess, John W. James, J. D. Kroeker, L. E. Seeley.

Nominating: C. B. Gamble, New Orleans, *Chairman;* B. W. Farnes, Portland, Ore., *Vice Chairman;* T. C. Caskey, Seattle, *Secretary;* Harry Horowitz, Albany; H. E. Russell, Cincinnati; M. W. Bishop, Chicago; H. B. Limbacher, Kalamazoo; T. H. Smoot, Marshalltown, Ia.; Fred Janssen, Denver; C. W. Leek, Vancouver, B. C.; H. G. S. Murray, Montreal, Que.; Alternates: W. A. Grant, Syracuse; G. T. Donceel, Oklahoma City.

Local Chapter Officers—1953

Arizona

Organized 1953
Headquarters, Phoenix

President.....V. J. Carns
Vice President.....K. E. Craft
Secretary.....A. J. Nelesen
Treasurer.....J. E. Scott
Board of Governors: A. C. Baechlin

Arkansas

Organized 1952
Headquarters, Little Rock

President.....Herman Cumnock
Vice President.....R. E. Blaylock
Secretary.....W. J. Franklin
Treasurer.....J. L. Brown

Atlanta

Organized 1937
Headquarters, Atlanta

President.....J. G. Croley
Vice President.....J. M. Lazenby
Secretary.....T. A. Barrow, Jr.
Treasurer.....Peter Van Dae, Jr.
Board of Governors: E. K. Jamison, F. A. Player

Baltimore

Organized 1949
Headquarters, Baltimore

President.....H. D. Glaser
Vice President.....E. H. Taze
Secretary.....G. J. Senner
Treasurer.....R. W. Pentecost
Board of Governors: E. S. Berngartt, R. T. Bollman, E. R. Kent

British Columbia

Organized 1952
Headquarters, Vancouver, B. C., Canada

President.....C. W. Leek
Vice President.....S. C. Gale
Secretary.....D. A. English
Treasurer.....Cornelius Van Boeyen
Board of Governors: D. M. Drake, R. D. Hale, D. B. Leaney, M. E. Minaker, C. H. Turland, S. W. Welsh

Central New York

Organized 1944
Headquarters, Syracuse

President.....C. R. Acheson
Vice President.....F. D. Putnam
Secretary.....H. K. Ormsby, Jr.
Treasurer.....L. A. Childs
Board of Governors: F. E. Krell, L. F. LaTart, Lawrence Macrow

Central Ohio

Organized 1944
Headquarters, Columbus

President.....N. T. Hess
Vice President.....J. A. Guy
Secretary.....R. A. Wilson
Treasurer.....W. A. Schoonover
Board of Governors: A. D. Bogen, B. W. Cornelius, B. E. Petry

Cincinnati

Organized 1932
Headquarters, Cincinnati

President.....R. C. Beineke
Vice President.....F. W. Wilson
Secretary.....A. H. Gerdsen
Treasurer.....R. G. Anderson
Board of Governors: T. D. Reiley, H. E. Russell

Connecticut

Organized 1940
Headquarters, New Haven

President.....G. F. Nieske
Vice President.....R. W. Sidbury
Secretary.....E. J. Hoagland
Treasurer.....C. L. L'Hommedieu
Board of Governors: T. L. Arnold, E. R. Clement, Walter Heywood

Delta

Organized 1939
Headquarters, New Orleans

President.....H. L. Salaun
Vice President.....C. V. Bankston
Secretary.....T. J. Jenny
Treasurer.....J. J. Friedler, Jr.
Board of Governors: L. R. Maxwell, H. N. Stall, Walter Cooke

Empire State Capital

Organized 1951
Headquarters, Albany

President.....N. W. Burrill
1st Vice President.....E. C. Doyle
2nd Vice President.....G. G. Davis
Secretary-Treasurer.....L. V. Appleby
Board of Governors: F. R. Foote, Harry Horowitz, H. F. Kruger, E. J. Mahoney

Golden Gate

Organized 1937
Headquarters, San Francisco

President.....J. E. Murray
Vice President.....Eric Roberts
Secretary.....D. E. McLeod
Treasurer.....F. K. Crouch
Board of Governors: H. L. Duncan, Edward Hill, Jr., T. J. Janes

Local Chapter Officers—1953 *(continued)*

Illinois

Organized 1906
Headquarters, Chicago

President.....H. G. Chapin
Vice President.....G. V. Zintel
Secretary.....H. G. Gragg
Treasurer.....G. G. Freyder
Board of Governors: M. W. Bishop, J. F. Cummiskey, Herbert Kreisman, C. R. Kuglin

Indiana

Organized 1943
Headquarters, Indianapolis

President.....R. C. Blackman
Vice President.....P. O. Patterson
Secretary.....J. W. Jackson
Treasurer.....A. O. Roche, Jr.
Board of Governors: W. W. Grear, J. T. Hardin, F. B. Morse

Inland Empire

Organized 1950
Headquarters, Spokane, Wash.

President.....G. M. Dieter
Vice President.....H. A. Bickel
Secretary.....F. W. Jenkinson
Treasurer.....Max Tonn
Board of Governors: G. C. Murray, T. P. Nau, R. D. Nevers

Iowa

Organized 1940
Headquarters, Des Moines

President.....J. F. Sandfort
Vice President.....L. E. Gnade
Secretary-Treasurer.....G. J. Kraai
Board of Governors: R. H. Schnell, H. A. Schwieger, W. A. Schworm

Kansas

Organized 1951
Headquarters, Wichita

President.....R. L. Pennington
Vice President.....Charles Yoe
Secretary.....O. B. Strahm
Treasurer.....T. L. Roberts
Board of Governors: R. F. Bauer, E. C. Blood, O. P. Bullock

Kansas City

Organized 1917
Headquarters, Kansas City, Mo.

President.....G. H. Stoffer
Vice President.....F. K. Ladewig
Secretary.....A. S. Hurt, Jr.
Treasurer.....R. M. Spencer
Board of Governors: H. A. Atwater, H. H. Lewis, W. A. Reichow

Manitoba

Organized 1935
Headquarters, Winnipeg, Man., Canada

President.....A. K. Piercy
Vice President.....J. A. Bell
Secretary-Treasurer.....D. B. Brace
Board of Governors: W. J. Atkinson, J. F. Bertram, G. E. Dahlgren, G. T. Christie, Ernest Lambert, A. J. McIntyre, H. F. Randall, J. C. Stangl

Massachusetts

Organized 1912
Headquarters, Boston

President.....A. L. Hare
Vice President.....G. W. Sprague
Secretary.....W. G. Martin, Jr.
Treasurer.....W. G. Burbo
Board of Governors: John Bonner, W. G. Burbo, A. M. Lovenberg, W. G. Martin, Jr., G. B. Torrens

Memphis

Organized 1944
Headquarters, Memphis

President.....A. W. Shelby
Vice President.....H. H. Wilson
Secretary.....A. T. Bevil
Treasurer.....L. V. Eberle, Jr.
Board of Governors: W. L. Drake, W. L. Henson, E. A. Smith, W. E. Thorpe

Miami Valley

Organized 1950
Headquarters, Dayton, Ohio

President.....J. M. Schweiger
Vice President.....R. W. Kimmel
Secretary.....D. E. Tullis
Treasurer.....J. B. Rishel
Board of Governors: L. P. Brehm, Jr., R. J. Perkins

Michigan

Organized 1916
Headquarters, Detroit

President.....R. H. Oberschulte
Vice President.....E. F. Glanz
Secretary.....D. S. Falk
Treasurer.....J. N. Livermore
Board of Governors: Axel Marin, D. L. McConachie, R. M. McIntosh, C. A. Strand

Minnesota

Organized 1918
Headquarters, Minneapolis

President.....V. E. Pearson
Vice President.....J. A. Craig
Secretary.....E. T. Erickson
Treasurer.....J. S. Locke
Board of Governors: J. G. Hamm, William Sturm, W. A. Swenberg

Local Chapter Officers—1953 *(continued)*

Mississippi

Organized 1953
Headquarters, Jackson

President.....Burt Lomax, Jr.
Vice President.....O. F. Rogers
Secretary.....J. G. Coleman, Jr.
Treasurer.....I. E. Rowe
Board of Governors: Robert Porter, J. W. Taylor, R. E. Vernon

Montreal

Organized 1936
Headquarters, Montreal, Que., Canada

President.....B. J. Horsburgh
Vice President.....D. L. Lindsay
Secretary.....R. J. Ker
Treasurer.....S. R. Plamondon
Board of Governors: Henri Dagenais, R. R. Duquette, W. G. Hole, G. N. Martin, H. G. S. Murray

Nebraska

Organized 1940
Headquarters, Omaha

President.....O. J. Smith
Vice President.....H. H. Ulrich
Secretary.....C. A. Failor
Treasurer.....L. J. Paulsen
Board of Governors: S. W. Black, C. A. Failor, L. J. Paulsen, O. J. Smith, H. H. Ulrich

New York

Organized 1911
Headquarters, New York

President.....J. E. Schechter
Vice President.....R. L. Stinard
Secretary.....C. H. Flink
Treasurer.....W. M. Heebner
Board of Governors: H. S. Birkett, Albert Giannini, P. B. Gordon, J. B. Hewett, Clifford Strock

North Jersey

Organized 1952
Headquarters, Newark

President.....F. H. Faust
Vice President.....H. M. Patrick
Secretary.....C. H. Smith
Treasurer.....W. C. Kruse, Jr.
Board of Governors: E. T. Best, M. C. Christesen, O. O. Oaks

North Texas

Organized 1938
Headquarters, Dallas

President.....P. N. Vinther
Vice President.....M. W. Brown
Secretary.....J. A. Ray
Treasurer.....H. G. Gregerson
Board of Governors: R. E. Allison, G. H. Mefert, J. L. Tye

Northeastern Oklahoma

Organized 1948
Headquarters, Tulsa

President.....J. N. Watt
Vice President.....V. W. Miles
Secretary.....J. C. Netherton, Jr.
Treasurer.....H. H. Hubbell
Board of Governors: T. W. Allen, C. H. Dollmeyer, Jr., R. W. Winget

Northern Ohio

Organized 1916
Headquarters, Cleveland

President.....J. M. Black
Vice President.....D. E. Mannen, Jr.
Secretary.....M. L. Erickson—Sherwood Nassau*
Treasurer.....W. F. D. Neiheiser
Board of Governors: L. C. Burkes, R. E. Sherman, J. R. Venning

Northern Piedmont

Organized 1952
Headquarters, Greensboro, N. C.

President.....J. E. Hart
Vice President.....R. B. Crosland, Jr.
Secretary.....S. T. Oliver
Treasurer.....L. R. Gorrell
Board of Governors: J. deBernard Shepard, L. L. Vaughan, R. A. White

Oklahoma

Organized 1935
Headquarters, Oklahoma City

President.....F. X. Loeffler, Jr.
Vice President.....E. F. Dawson
Secretary.....G. E. Ervin
Treasurer.....F. R. Denham
Board of Governors: J. H. Carnahan, W. J. Collins, Jr., G. T. Donceel

Ontario

Organized 1922
Headquarters, Toronto, Ont., Canada

President.....N. W. Kingsland
Vice President.....M. C. Bailey
Secretary-Treasurer.....H. R. Roth
Board of Governors: D. L. Angus, C. J. Bootes, J. H. Ross, Charles Torry

Oregon

Organized 1939
Headquarters, Portland

President.....A. N. Hoss
Vice President.....E. E. Kelly
Secretary.....F. E. Finkbeiner
Treasurer.....Dick Blankenship
Board of Governors: G. R. Folds, Jr., K. H. Hanson, W. B. Hayes

---

* Filled unexpired term.

Local Chapter Officers—1953 *(continued)*

Ottawa Valley

Organized 1952
Headquarters, Ottawa, Ont., Canada

President.....G. L. Ostiguy
Vice President.....W. J. Robinson
Secretary.....N. J. Howes
Treasurer.....E. J. Schoenherr
Board of Governors: G. R. Hamilton, J. S. Massiah, W. B. Pennock, Bernard Stotesbury

Pacific Northwest

Organized 1928
Headquarters, Seattle, Wash.

President.....H. T. Orebaugh
1st Vice President.....T. C. Caskey
2nd Vice President.....I. L. Winsor
Secretary.....L. F. Christofferson
Treasurer.....E. W. Triol
Board of Governors: R. C. Lewis, M. W. McKinstry, W. B. Pride

Philadelphia

Organized 1916
Headquarters, Philadelphia

President.....C. F. Dietz
1st Vice President.....L. M. Church
2nd Vice President.....A. M. Robertson
Secretary.....P. H. Yeomans
Treasurer.....C. J. Lubking
Board of Governors: M. E. Barnard, L. M. Church, C. F. Dietz, C. J. Forve, C. J. Lubking, A. M. Robertson, P. H. Yeomans

Pittsburgh

Organized 1919
Headquarters, Pittsburgh

President.....C. H. Schneider
Vice President.....E. C. Hach
Secretary.....E. H. Riesmeyer, Jr.
Treasurer.....C. L. Benn
Board of Governors: H. B. Grabman, B. B. Reilly, O. L. Williams

Rocky Mountain

Organized 1944
Headquarters, Denver, Colo.

President.....H. L. Wray
Vice President.....V. F. Vallero
Secretary.....J. F. Cipra, Jr.
Treasurer.....Leo Krisl
Board of Governors: John Berngen, N. H. Brickham, H. H. Smith

Sacramento Valley

Organized 1952
Headquarters, Sacramento, Calif.

President.....E. C. McKinsey
Vice President.....J. E. Marshall
Secretary.....R. A. Sarro
Treasurer.....V. W. Thornburg
Board of Governors: J. E. Day, N. W. Downes, L. A. O'Meara

St. Louis

Organized 1918
Headquarters, St. Louis

President.....H. C. Sharp
1st Vice President.....G. H. Bemarkt
2nd Vice President.....J. F. Naylor, Jr.—E. T. Clucas*
Secretary.....W. P. Norris
Treasurer.....F. E. Ince
Board of Governors: C. H. Burnap, D. E. Hanlon, N. J. Hubbich, Jr., J. B. Killebrew

Shreveport

Organized 1948
Headquarters, Shreveport, La.

President.....M. A. Hudson
Vice President.....J. S. Malahy, Jr.
Secretary.....R. S. Segall
Treasurer.....H. E. Scott
Board of Governors: J. L. Collins, Jr., W. S. Evans, R. M. Hood

South Texas

Organized 1938
Headquarters, Houston

President.....E. G. Floeter, Jr.
Vice President.....J. W. Holland
Secretary.....F. M. Neil
Treasurer.....J. C. Lewis
Board of Governors: C. L. Fleming, O. G. Rivoire, A. B. Ullrich, Jr.

Southern California

Organized 1930
Headquarters, Los Angeles

President.....L. B. Davenport
Vice President.....C. D. Walz
Secretary.....J. L. McCullough
Treasurer.....H. F. Ulovec
Board of Governors: J. S. Earhart, F. B. Gardner, L. R. Kelly, T. A. Marshall, E. D. Pierce

Southern Piedmont

Organized 1952
Headquarters, Charlotte, N. C.

President.....R. E. Mason
Vice President.....R. S. Fullerton
Secretary.....S. B. Blanton
Treasurer.....W. E. Stowe, Jr.
Board of Governors: R. M. Warren, Jr., W. P. Wells, C. N. Witmer

Southwest Texas

Organized 1946
Headquarters, San Antonio

President.....D. E. Locher
Vice President.....L. H. Hornor, Jr.
Secretary-Treasurer.....F. B. Frazee
Board of Governors: F. C. Benham, F. B. Frazee, L. H. Hornor, Jr., D. E. Locher, W. E. Long, L. S. Pawkett

---

* Filled unexpired term.

Local Chapter Officers—1953 *(continued)*

Utah

Organized 1944
Headquarters, Salt Lake City

President.....M. L. Gollaher
Vice President.....Alfred Richeda
Secretary-Treasurer.....G. L. Soderborg, Sr.

Virginia

Organized 1946
Headquarters, Norfolk

President.....E. D. Duval
Vice President.....H. E. Marquart
Secretary.....T. B. Carpenter
Treasurer.....F. R. Ervin
Board of Governors: R. L. Burton, Jr., C. G. Conaway, Jr., D. E. Phillips

Washington, D. C.

Organized 1935
Headquarters, Washington, D. C.

President.....J. G. Muirheid
Vice President.....G. C. F. Asker
Secretary.....W. C. Reamy, Jr.
Treasurer.....J. W. Alexander
Board of Governors: J. C. Benson, J. J. Nolan, Jr., G. R. Walz

West Texas

Organized 1953
Headquarters, Lubbock

President.....R. A. Mixon
Vice President.....L. C. McKay
Secretary.....R. B. Carow
Treasurer.....J. C. Wharton
Board of Governors: M. J. Aderton, C. P. Houston, H. L. Mayes

Western Michigan

Organized 1931
Headquarters, Grand Rapids

President.....J. T. Anderson
Vice President.....F. W. Brundage
Secretary.....R. J. Waalkes
Treasurer.....S. M. Paganelli
Board of Governors: N. A. Buckley, H. R. Limbacher, R. L. Wells

Western New York

Organized 1919
Headquarters, Buffalo

President.....C. W. Stone
1st Vice President.....R. E. Lang
2nd Vice President.....Q. P. Thompson
Secretary.....C. W. Kaupp
Treasurer.....M. C. Beman
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, D. J. Mahoney, J. M. Quackenbush

Wisconsin

Organized 1922
Headquarters, Milwaukee

President.....N. E. Hill
Vice President.....L. C. Plaehn
Secretary.....H. K. Forfar
Treasurer.....I. J. Rossiter
Board of Governors: J. A. Lofte, H. H. Valiquet, J. R. Vernon

Special Branch

Switzerland

Organized 1952
Headquarters, Zurich

President.....L. F. DeBruyn
Vice President.....Walter Sennhauser
Secretary.....Walter Hausler
Treasurer.....Hans Frei

Student Branches

North Carolina State College

Organized 1948
Headquarters, Raleigh

President.....Norman Falbaum
Vice President.....A. D. Ehrman
Secretary.....J. L. Flowers
Treasurer.....F. O. Drummond, Jr.
Reporter.....J. C. Deal

Oklahoma A. & M. College

Organized 1950
Headquarters, Stillwater

President.....S. N. Cohen..Elliott Todd, Jr.
Vice President.....J. H. Churchwell
Secretary.....H. G. Rice..R. P. Schmidt
Treasurer.....V. O. Wagner
Student Council Representative.....T. M. Carr

Local Chapter Officers—1953 *(continued)*

Oregon State College

Organized 1949
Headquarters, Corvallis

President.....David Marx
Vice President.....John Malarkey
Secretary.....W. E. Norman
Treasurer.....R. G. Potter

Purdue University

Organized 1948
Headquarters, W. Lafayette, Ind.

Chairman.....R. G. Supple
Secretary.....R. E. Jacobs

Texas A. & M. College

Organized 1946
Headquarters, College Station

University of Detroit

Organized 1949
Headquarters, Detroit, Mich.

President.....J. R. Cairns
Vice President.....N. L. Novak
Secretary.....W. E. Zimmerman
Recording Secretary.....J. C. Rumpf

University of Kansas

Organized 1949
Headquarters, Lawrence

University of Texas

Organized 1949
Headquarters, Austin

University of Toronto

Organized 1951
Headquarters, Toronto, Ont., Canada

President.....D. S. Harper
Secretary.....L. S. Mitchell

1462

TRANSACTIONS

of

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1462

FIFTY-NINTH ANNUAL MEETING 1953

CHICAGO, ILLINOIS

FIRST SESSION, MONDAY, JANUARY 26, 9:30 A.M.

THE 59th Annual Meeting was called to order in the Grand Ballroom of the Conrad Hilton Hotel, Chicago, by First Vice Pres. Reg. F. Taylor owing to the illness of Pres. Ernest Szekely. It was the largest Society meeting ever held with a total attendance of 2814, including 1620 members, 363 ladies, and 831 guests.

REPORT OF COUNCIL

As required by Section 46 of the Membership Corporations' Law of the State of New York the following report is presented by the Council and filed herewith:

Assets

The Society has the following Assets:

Land and Laboratory Buildings at Cleveland, Ohio	$ 81,012.89
Furniture and Equipment, Tools, etc. at New York and Cleveland	42,106.80
On Deposit in New York and Cleveland banks	118,929.31
Securities (United States and Canadian Government Bonds) with Bankers Trust Company, New York as custodian	105,070.63
Accounts Receivable	14,191.27
Inventory Items	12,669.27
Deferred Items	8,028.84
Miscellaneous	2,911.40
TOTALS	$384,920.41
1951 TOTAL	395,803.37
DECREASE	$ 10,882.96

The sources of Society Cash Receipts were: Admissions Fees and Dues $122,425.80, Publications $156,772.67, Interest $1,665.42, From Research $175,593.51; and Miscellaneous $442.50—Total $456,899.90. Cash Disbursements were: Meetings, Committees and Chapters $26,142.20, Publications $125,606.93, Headquarters $116,693.29, Fund Raising $13,734.89, and Research $169,050.24—Total $451,227.55.

The Society's Reserve Fund was increased by the addition of $7,243.00 Admission Fees and $1,995.49 Interest earned on Savings, Bank Accounts and Securities. This is an increase of $9,238.49, bringing the Reserve Fund to $84,442.89.

Now in the Custodian Account of the Bankers Trust Co., New York, there are securities of the United States and Canadian Governments and a utility which cost: General Fund $35,790.61, Reserve Fund $53,631.50, and F. Paul Anderson Fund $1,000.00. The Research Reserve Fund now stands at $10,338.30.

Liabilities

The Society has the following Liabilities:

Reserve for building maintenance	$ 2,000.00
Accounts Payable	—0—
Federal Withholding Taxes	1,275.85
Accrued Accounts	11,440.13
Deferred Income:	
Research Projects	6,274.39
Prepaid Dues, Initiation Fees, etc.	4,597.96
Reserve for TRANSACTIONS	15,341.91
Net Worth (Funds)	343,990.17
TOTAL	$384,920.41

At the end of the fiscal year, October 31, 1952, the Balance Sheet shows the Society's Net Worth as $343,990.17 compared to $358,041.65 on October 31, 1951, a net loss of $14,051.48.

The Property Fund covering the Laboratory Land and Buildings at Cleveland, Laboratory Equipment and Tools, Furniture and Fixtures at Headquarters, New York and Cleveland is $123,119.69.

Fire insurance of $200,000.00 is carried on buildings, $137,500.00 on contents, sprinkler $12,000.00, also Comprehensive Bodily Injury $100,000-$300,000 and Auto Property Damage $5,000.00, Medical, $1,000.00. Fidelity Bonds on Officers and all employees $20,000.00. Workmen's Compensation is carried in New York and Ohio, as required by law.

The Society has contributed to the Employees Retirement Plan Trust $2,313.80 and the participating employees contributed a similar amount deducted from salary.

Total cash in New York and Cleveland banks $118,929.31, consisting of $61,623.73 in General Operating Fund, $19,841.12 in Reserve Fund, $195.91 in F. Paul Anderson Fund, $25,689.62 in the Research Fund, $10,338.30 in Research Reserve, $704.10 in Research Endowment, and $536.82 in the Building Fund.

Membership

The Society admitted the following members in the grades indicated since January 1, 1952:

Members	283
Associates	10
Affiliates	316
Junior Members	339
Students	130
TOTAL	1078

The names and addresses of the candidates for membership were published in the JOURNAL of the Society each month during the year and are on record in the Secretary's office. The present membership total is 8,468.

Respectfully submitted,
ERNEST SZEKELY, *President*
J. DONALD KROEKER, *Treasurer*

Treas. J. Donald Kroeker presented the Treasurer's Report, including the audit of the Society's accounts by Frank G. Tusa & Co.

TREASURER'S REPORT

In conformity with the By-Laws, the audit of the Society's accounts is presented herewith in the form of a *Report of Examination* for the fiscal year ending October 31, 1952, prepared by Frank G. Tusa & Co., Certified Public Accountants.

Viewing the operation of the Society for the fiscal year reported, certain factors may be of interest. On an accrual basis, expenses are shown exceeding income by $25,863.27. On a cash basis, receipts exceeded disbursements by $5,672.35. The Council was faced this year with several natural conditions:

a. While dues levels were fixed, it was necessary to increase staff for service to a membership which has nearly trebled in the last ten years.

b. Research income was below normal. The year was the first in six years without the substantial research income from the International Exposition Co. in its operation of the heating and ventilating expositions. Further, research income was affected by implementation of a new type of program for fund raising, requiring time and experience for full effectiveness.

So the year was in a measure one of adjustment, in which the Council found that, to conduct the Society's fiscal program wisely, it was necessary to dip into reserves provided for the purpose.

And yet the Society's financial structure remains in excellent condition, as figures on net worth show. The growth of net worth during the last 22 years shows that the net worth has increased from $23.50 per member 20 years ago to $40.00 this year, more than two and a half times the amount of unearmarked dues per member.

For every dollar of unearmarked dues and admission fees the Society had an income of $3.76.

Several actions of the Council during the fiscal year are significant:

a. A building fund of some $35,000 was created by setting aside certain securities for availability in the event it becomes feasible to erect quarters for the headquarters operation.

b. An Investment Advisory Committee has been created. This committee, composed of prominent business men of the Society, serving for extended terms, will advise the Finance Committee on investments in a continuity toward a planned investment program.

c. Accounting has been changed to an accrual basis better to reflect fiscal status.

During the year the Finance Committee developed a study of the effect of increased membership, which has pointed toward desirability, if not necessity, of an active membership encouragement program, if dues are held at current levels. The last such program was pursued in 1949. A net increase of 1000 members per year is considered attainable with parallel improvement in quality.

The Society's structure for conducting the basic and fundamental research, to which it is dedicated and the backlog of which is tremendous, is suitable to pursue an annual program extending toward a million dollars. The chief limitation is the funds available.

Here is a point at which self evaluation toward more general participation is indicated. It is conservatively estimated that more than $20,000,000 is expended annually in research, including that toward product development, in the field of heating, ventilat-

ing and air conditioning. Of this amount, less than one percent is expended exclusively toward basic and fundamental research by the Society dedicated to this activity.

No one can question the very material progress in the art of heating, ventilating, and air conditioning which has resulted from the Society's research. Evidence and proof are at every hand. It is therefore the base upon which each of our businesses and professions rests. And yet there are relatively very few companies and virtually no professional offices which contribute financially toward research to strengthen and enlarge this base.

It would seem logical and certainly painless for each business and professional office to contribute *regularly* to the research program according to its own formula, perhaps based on a fractional percentage of its net income. I suggest this to you for your serious consideration.

If this is done, the list of contributors could easily swell from about 125 to 3000 within a few years, the research base upon which our works rests would be firmed and broadened, and we would see the art advance toward greater benefits to ourselves and the public.

In conclusion, it is obvious your Society is in good financial condition; has seen remarkable growth, and should be expected to have a greater and more beneficial future.

Respectfully submitted,
J. DONALD KROEKER, *Treasurer*

Accountant's Report

FRANK G. TUSA & CO.
CERTIFIED PUBLIC ACCOUNTANTS
37 Wall St., New York 5, N. Y.

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
62 Worth Street,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and Cleveland and its related funds for the fiscal year ended October 31, 1952, and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1952. For the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements, and the disbursements were supported by payment vouchers. The dues from members and the interest from investments were accounted for.

A balance sheet reflecting the financial condition of the Society as of the close of business October 31, 1952 is submitted herewith and your attention is directed to the following comments thereon:

CASH

The cash on deposit was verified by direct communication with commercial and savings banks and the balances reported to us were reconciled with those reflected by the books of the Society. A schedule of cash is included as a part of this report.

Checks representing the cash on hand for deposit were inspected by us and the cash on hand was counted.

MARKETABLE SECURITIES

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Co., where same are deposited for safe-keeping. Securities have been included in the accompanying balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

Accounts Receivable

Trial balances taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1952 were classified and aged as follows:

Membership	1951	1952	Total
Members	$ 17.16	$ 3,209.81	$ 3,226.97
Associates	161.75	4,427.48	4,589.23
Affiliates	—0—	1,098.25	1,098.25
Juniors	5.00	2,182.72	2,187.72
Students	6.00	63.00	69.00
Totals	$189.91	$10,981.26	$11,171.17

Sundry Debtors

Unpaid Charges made during 1952	$4,678.71
Unpaid Charges made during 1951	184.37
Unpaid Charges made during 1950	742.04
Total	$5,605.12

After writing off all receivables known to be uncollectible, it is our opinion that the reserves for dues and accounts receivable doubtful of collection reflected in the accompanying balance sheet are ample to cover collection losses that may be incurred.

Research Fund Account Receivable

The balance due to the Research Fund from the Society as at October 31, 1952 represents the balance of 40 percent of dues receivable from Members, Associates and Affiliates as of such date.

Inventories

The following inventories were verified either by count or direct communication:

Transactions	$ 5,413.57
Emblems	359.84
Guide Paper	6,670.86
Membership Roll Paper	225.00
Total	$12,669.27

A schedule of Transactions inventoried follows:

Volume	Year	Quantity	Price	Amount
1-49	1895-1943	3,266	$ .40	$1,306.40
50	1944	169	1.82	307.58
51	1945	335	1.91	639.85
52	1946	410	1.67	684.70
53	1947	144	1.77	254.88
54	1948	782	1.78	1,391.96
55	1949	410	2.02	828.20
Total				$5,413.57

Permanent Assets

The land and buildings, instruments, equipment and furniture and fixtures are reflected on the balance sheet at cost of acquisition. With the exception of land and buildings all assets have been depreciated at the rate of 10 percent per Annum. During the year fully depreciated furniture and fixtures in the amount of $37.88 was written off against the reserve.

In accordance with the resolution adopted by the Council at its meeting of January 23, 1949 depreciation was not provided on the buildings for the current fiscal year.

TAXES

The sum of $1,275.85 represents Federal income taxes withheld from employees salaries during the month of October, 1952.

DEFERRED INCOME

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follows:

MEMBERSHIP	AMOUNT
Members	$ 358.04
Associates	70.37
Affiliates	209.25
Juniors	563.30
Students	43.50
TOTAL	$1,244.46

RESERVE FOR TRANSACTIONS

Out of the original $12,500.00 reserve for TRANSACTIONS Volume 56 there remains an unexpended balance of $3,341.91. The reserve of $12,000.00 provided for Volume 57 in the 1952 budget is also reflected in the accompanying balance sheet. It is recommended that the amount in this reserve be increased inasmuch as the printing and distribution of Volume 55 exceeded $16,000.00.

RESERVE FOR BUILDING MAINTENANCE

On October 11, 1952 the Council voted:

"That disbursements be made from the Building and Maintenance Reserve Fund as authorized by the Building Committee for major items of repair and maintenance, but any balance remaining in the fund beyond the expenditures for each year would go into reserve until the Building Maintenance Reserve Fund reached a total of $10,000.00."

Inasmuch as there were no disbursements authorized as at the end of the current fiscal year, there exists a reserve balance of $2,000.00 as at October 31, 1952.

FUNDS

There is included as a part of this report an analysis of funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1952.

On January 27, 1952 the Council created a Building Fund which is reflected on the balance sheet submitted as a part of this report.

DEPOSITS RECEIVABLE

The deposit placed with the United Airlines in the sum of $425.00 was verified by direct communication.

ADVANCES

The indebtedness from employees in New York and Cleveland represents advances to the employees retirement plan in the sums of $1,509.43 and $976.97 respectively. This indebtedness is to be reimbursed by the trust during the ensuing fiscal year.

DEFERRED CHARGES

The following prepayments have been deferred to future expenses:

H.P.A.C. Subscription	$2,717.55
Advertising Sales Promotion	1,350.00
Insurance Premiums—New York	517.92
Insurance Premiums—Cleveland	3,443.37
TOTAL	$8,028.84

INSURANCE

The insurance coverages of the Society follow:

FIRE		
Building (Cleveland)		$200,000.00
Personal Property:		
Cleveland	$100,000.00	
New York	25,500.00	
Printers	12,000.00	137,500.00
SPRINKLER LEAKAGE		
Waverly Press	10,000.00	
Hawes and Petit	2,000.00	12,000.00
NON-OWNERSHIP—AUTOMOBILE—COMPREHENSIVE		
Public Liability		100/300M
Property Damage		5M
Medical		1M
GENERAL PUBLIC LIABILITY—PREMISES		
New York and Cleveland		
Liability		100/300M
Bodily Injury		25/100M
COMMERCIAL BLANKET BOND		
All employees including the President, Treasurer and Chairman of the Finance Committee		20,000.00

Respectfully submitted,

FRANK G. TUSA & CO.
Certified Public Accountants

BALANCE SHEET

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New York, N. Y. — October 31, 1952

ASSETS

GENERAL FUNDS			
SOCIETY ASSETS			
CASH			
On Hand and on Deposit			$61,623.73
SECURITIES			
At Cost (Market Value $3,705.00)		$3,782.36	
Add: Accrued Interest		27.50	3,809.86
ACCOUNTS RECEIVABLE			
Membership Dues	$11,171.17		
Less: Reserve for Doubtful	6,702.70	4,468.47	
Advertisers and Sundry Debtors	5,605.12		
Less: Reserve for Doubtful	1,000.00	4,605.12	
Initiation Fees		55.00	9,128.59
INVENTORIES			12,669.27
DEPOSIT WITH UNITED AIRLINES			425.00
ADVANCES TO EMPLOYEES PENSION TRUST			1,509.43
DEFERRED CHARGES			4,585.47
(Forward)			$ 93,751.35

(Forwarded)				$ 93,751.35
RESEARCH ASSETS				
CASH				
On Hand and on Deposit			25,689.62	
ACCOUNTS RECEIVABLE				
40% of Dues	4,206.41			
Less: Reserve for Doubtful	2,139.47	2,066.94		
Due from U. S. Navy Department		5,037.83		
Hospitalization—Employees		9.85		
Miscellaneous		15.00	7,129.62	
ADVANCES TO EMPLOYEES PENSION TRUST			976.97	
DEFERRED CHARGES			3,443.37	$ 37,239.58
PROPERTY FUND				
Land and Buildings		$84,920.96		
Less: Reserve for Depreciation		3,908.07	$81,012.89	
Laboratory and Equipment (Cleveland)		62,816.01		
Less: Reserve for Depreciation		29,934.06	32,881.95	
Furniture and Fixtures (New York)		12,386.34		
Less: Reserve for Depreciation		3,761.49	8,624.85	
Library (Cleveland)			300.00	
Tools (Cleveland)			300.00	123,119.69
BUILDING FUND				
CASH				
On Hand and on Deposit			536.53	
SECURITIES				
At Cost (Market Value $36,110.20)		32,008.25		
Add: Accumulated and Accrued Interest		3,638.25	35,646.50	36,183.03
RESERVE FUND				
Cash on Deposit			19,841.12	
Securities at Cost (Market Value $64,601.77)		53,631.50		
Add: Accumulated Interest		10,970.27	64,601.77	84,442.89
ENDOWMENT FUNDS				
F. PAUL ANDERSON FUND				
Cash on Deposit			195.91	
Securities at Cost (Market Value $964.00)		1,000.00		
Add: Accrued Interest		12.50	1,012.50	1,208.41
RESEARCH FUND				
Cash on Deposit				704.10
RESEARCH RESERVE FUND				
Cash on Deposit				10,338.30
				$386,987.35

LIABILITIES

GENERAL FUND				
FEDERAL WITHHOLDING TAXES			$ 1,275.85	
DUE RESEARCH			2,066.94	
ACCRUED ACCOUNTS				
Salaries and Commissions		$10,440.13		
Professional Fees		1,000.00	11,440.13	
DEFERRED INCOME				
Prepaid Membership Dues:				
Elected Members	$1,244.46			
Candidates	1,003.50	2,247.96		
Prepaid Admission Fees		2,250.00		
Junior Student Member Award		100.00	4,597.96	
RESERVE FOR PUBLICATIONS				
TRANSACTIONS—Volume 56		3,341.91		
TRANSACTIONS—Volume 57		12,000.00	15,341.91	$ 34,722.79
RESEARCH FUND				
RESERVE FOR BUILDING MAINTENANCE			2,000.00	
DEFERRED INCOME				
Sound Energy Studies		574.39		
Periodic Heat Flow Through Roofs		1,850.00		
Summer Infiltration		1,850.00		
Sorption		2,000.00	6,274.39	8,274.39

FUNDS

Society General Fund	59,028.56	
Research General Fund	28,965.19	
Property Fund	123,119.69	
Building Fund	36,183.03	
Society Reserve Fund	84,442.89	
F. Paul Anderson Fund	1,208.41	
Research Endowment Fund	704.10	
Research Reserve Fund	10,338.30	343,990.17
		$386,987.35

STATEMENT OF INCOME AND EXPENSES

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.

For the Fiscal Year Ended October 31, 1952

INCOME

INCOME FROM DUES			
104—MEMBERS	$ 83,685.50		
Less: Cancellations	1,488.25	$ 82,197.25	
Less: 40% to Research Fund		32,878.90	$ 49,318.35
105—ASSOCIATES	57,131.25		
Less: Cancellations	3,095.75	54,035.50	
Less: 40% to Research Fund		21,614.20	32,421.30

105A—AFFILIATES	14,125.00			
Less: Cancellations	193.75	13,931.25		
Less: 40% to Research Fund		5,572.50	8,358.75	
106—JUNIORS		24,944.50		
Less: Cancellations		1,260.75	23,683.75	
107—STUDENTS		231.00		
Less: Cancellations		448.50	(217.50)	
TOTAL DUES				$113,564.65
OTHER INCOME				
108—Admission Fees			7,243.00	
109—Emblems			626.88	7,869.88
INCOME FROM PUBLICATIONS				
115—JOURNAL Contract			23,500.00	
300—GUIDE			126,846.41	
117—TRANSACTIONS			1,159.25	
118—Books, Reprints and Codes			3,016.95	154,522.61
INVESTMENT INCOME				
125—Interest—Savings Banks			765.17	
126—Interest—Securities			140.69	905.86
RESEARCH				
FROM A.S.H.V.E. 40% of Members, Associates and Affiliates Dues				$ 60,065.60
FROM U. S. NAVY RESEARCH				27,061.95
CONTRIBUTIONS				
GENERAL			$ 19,272.94	
EARMARKED				
Received during the current fiscal year —Per Schedule	$ 45,321.84			
Add: Deferred from prior years	3,924.39	$ 49,246.23		
Deduct: Deferred to future operations		6,274.39	42,971.84	62,244.78
INTEREST EARNED				1.48
				149,373.81
TOTAL INCOME				$426,236.81

EXPENSES

COMMITTEES AND CHAPTERS—SCHEDULE F			
150—Officers Travel	$ 2,728.52		
151—Council and Council Committee Travel	7,388.02		
164—Standards Committee Travel	55.88		
170—Admissions and Advancement Committee Travel	625.15		
172—Nominating Committee Travel	1,431.18		
173A—Chapter Speakers Travel	2,308.75		
173B—Chapters Conference Committee Travel	3,647.04		
173C—Chapter Relations Committee	594.37		
201—ASA Membership Dues	100.00		
204—Membership Certificates	1,340.30		
205—Emblems	781.09		
206—Medals and Awards	321.49	21,321.79	

MEETINGS—SCHEDULE G

163—Meetings	4,213.83	
327—Chapter Annual and Semi-Annual Meetings Allowances	787.47	5,001.30

PUBLICATIONS—SCHEDULE H

200—Members Subscriptions to HPAC	15,876.54	
202—TRANSACTIONS	15,006.26	
203—Membership Roll	8,888.08	
208—Books, Reprints, Codes, Charts, etc.	2,499.54	
305—THE GUIDE	81,201.60	123,472.02

HEADQUARTERS—SCHEDULE I

210—Salaries—Secretary and Staff	71,132.46	
210A—Social Security Taxes	897.85	
211—Public Relations	4,842.00	
212—Traveling—Secretary and Staff	2,581.06	
213—Rent and Light	11,254.98	
214—Telephone	2,008.66	
215—Telegraph	615.12	
216—Postage	5,279.84	
217—Printing and Stationery	6,622.69	
219—Addressing and Address Changes	366.27	
220—Professional Services	3,363.98	
221—Bank Charges	49.12	
222—Insurance	236.67	
11—Depreciation—Furniture and Fixtures	1,127.11	
223—Office Expenses	2,796.67	
224—Pension	1,096.51	114,270.99

FUND RAISING—SCHEDULE K

350—Salaries and Fees	12,012.93	
351—Travel	594.75	
352—Printing	897.58	
354—Postage and Miscellaneous	229.63	13,734.89

RESEARCH—SCHEDULE J

COMMITTEE EXPENSES

Committee Travel	2,830.00	
Chairman and Executive Committee Travel	1,439.59	4,269.59

STAFF SALARIES

Administrative, Engineering and Technical	51,812.48	
Clerical	12,360.00	
Pensions	1,217.29	
Part Time	13,382.03	
Consulting Services	925.67	
Social Security	1,074.12	80,771.59

LABORATORY EXPENSES

General Staff Travel	3,271.42	
Society Meeting Travel	1,982.99	
Postage	536.80	
Telephone and Telegraph	1,741.65	
Office Supplies and Printing	942.40	
Mimeographing and Blue Prints	2,222.68	
Library and Reprints	491.35	
Professional Services	438.72	
Workmen's Compensation Insurance	257.98	
Laboratory Materials and Supplies	4,181.88	
Depreciation of Equipment	5,936.52	22,004.39

BUILDING OPERATIONS AND MAINTENANCE			
Power	2,725.45		
Heating	1,393.37		
Water	29.24		
Janitors Wages	2,817.69		
Janitors Supplies	305.56		
Building Maintenance	789.88		
Building Insurance	1,451.14	9,512.33	
PROVISION FOR BUILDING MAINTENANCE RESERVE		2,000.00	
U. S. NAVY RESEARCH			
Staff Salaries	12,665.78		
Staff Travel	281.39		
Materials and Supplies	2,524.87		
Equipment	2,367.79	17,839.83	
COOPERATIVE RESEARCH AGREEMENTS			
Case Institute of Technology	2,500.00		
Columbia University	500.00		
Kansas State College	3,200.00		
Northwestern University	4,000.00		
University of Florida	1,500.00		
University of Illinois	6,500.00		
University of Minnesota	4,454.52		
California Institute of Technology	3,000.00		
Michigan State College	800.00		
University of Pittsburgh	2,000.00	28,454.52	164,852.25
		165,474.47	443,275.46
NET LOSS			(17,038.65)
OTHER DEDUCTIONS			
Admission Fees Allocated to Reserve Fund		7,243.00	
Uncollectible Accounts Receivable Written Off		440.99	
Uncollectible Dues Written Off		1,811.34	
		9,495.33	
OTHER INCOME			
Income from Fluctuation Canadian Exchange		670.71	8,824.62
EXCESS OF EXPENSES OVER INCOME			($25,863.27)

BUDGET COMPARISON — CASH RECEIPTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New York, N. Y. — For the Fiscal Year Ended October 31, 1952

	Actual	Budget Provision	Increases	Decreases
DUES—SCHEDULE A				
CURRENT MEMBERS				
6A—Members	$ 44,665.90	$ 43,200.00	$ 1,465.90	
6B—Associates and Affiliates	36,402.93	40,500.00		$ 4,097.07
6C—Juniors	18,483.15	19,500.00		1,016.85
6D—Students	481.40	800.00		318.60
	100,033.38	104,000.00	1,465.90	5,432.52

New Members				
104—Members	2,773.50	1,800.00	973.50	
105—Associates and Affiliates	3,030.30	2,100.00	930.30	
106—Juniors	3,512.04	2,000.00	1,512.04	
107—Students	163.00	150.00	13.00	
	9,478.84	6,050.00	3,428.84	
Prior Years Dues				
6E—Members	1,232.46	1,800.00		567.54
6F—Associates and Affiliates	1,765.31	2,700.00		934.69
6G—Juniors	555.97	500.00	55.97	
6H—Students	144.00	100.00	44.00	
	3,697.74	5,100.00	99.97	1,502.23
Total Dues	113,209.96	115,150.00	4,994.71	6,934.75
108—Admission Fees	5,615.00	5,500.00	115.00	
108A—Admission Fees Prepaid	2,180.00	250.00	1,930.00	
	7,795.00	5,750.00	2,045.00	
Deferred Income				
6I—Prepaid Dues by Members	494.34	200.00	294.34	
6J—Prepaid Dues by Candidates	926.50	800.00	126.50	
	1,420.84	1,000.00	420.84	
Publication—Schedule B				
115—Journal Contract	23,500.00	23,500.00		
300—Guide Advertising	78,146.45	75,800.00	2,346.45	
302—Guide Copy	50,892.09	53,700.00		2,807.91
117—Transactions	878.75	1,750.00		871.25
118—Books, Reprints, Codes, etc.	3,355.38	1,750.00	1,605.38	
	156,772.67	156,500.00	3,951.83	3,679.16
Investments—Schedule C				
125—Interest—Savings Banks	990.70	800.00	190.70	
126—Interest—Securities	674.72	800.00		125.28
	1,665.42	1,600.00	190.70	125.28
Other Receipts—Schedule D				
109—Emblems	442.50	500.00		57.50
Research Laboratory Receipts—Schedule E				
40% of Members, Associates and Affiliates Dues	59,913.58	61,400.00		1,486.42
Contributions—General	19,272.94	15,000.00	4,272.94	
Contributions—Earmarked	45,321.84	45,000.00	321.84	
Navy Contract	25,470.36	34,260.00		8,789.64
Interest	614.79	250.00	364.79	
From Reserve	25,000.00	25,000.00		
	175,593.51	180,910.00	4,959.57	10,276.06
Total Receipts	$456,899.90	$461,410.00	$16,562.65	$ 21,072.75

BUDGET COMPARISON — CASH DISBURSEMENTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

New York, N. Y. — For the Fiscal Year Ended October 31, 1952

	Actual	Budget Provision	Increases	Decreases
COMMITTEES AND CHAPTERS—SCHEDULE F				
150—Officers Travel	$ 2,728.52	$ 4,000.00	$	$ 1,271.48
151—Council and Council Committee Travel	7,388.02	8,000.00		611.98
164—Standards Committee Travel	55.88	750.00		694.12
170—Admission and Advancement Committee Travel	625.15	600.00	25.15	
171—Charter and By-Laws Committee Travel	—0—	500.00		500.00
172—Nominating Committee Travel	1,431.18	2,000.00		568.82
173A—Chapter Speakers Travel	2,308.75	2,500.00		191.25
173B—Chapters Conference Committee Travel	3,647.04	5,000.00		1,352.96
173C—Chapter Relations Committee	594.37	600.00		5.63
175—Building Committee Travel	—0—	300.00		300.00
201—*A.S.A.* Membership Dues	100.00	100.00		
204—Membership Certificates	1,340.30	700.00	640.30	
205—Emblems	625.99	250.00	375.99	
206—Medals and Awards	295.70	350.00		54.30
	21,140.90	25,650.00	1,041.44	5,550.54
MEETINGS—SCHEDULE G				
163—Meetings	4,213.83	4,000.00	213.83	
327—Chapter Meeting Allowance	787.47	2,000.00		1,212.53
	5,001.30	6,000.00	213.83	1,212.53
PUBLICATIONS—SCHEDULE H				
200—JOURNAL Subscriptions	16,003.59	17,000.00		996.41
202—TRANSACTIONS—Volumes 55-57	17,479.72	12,000.00	5,479.72	
203—Membership Roll	9,113.08	7,200.00	1,913.08	
208—Books, Reprints, Codes, etc.	2,499.54	1,500.00	999.54	
THE GUIDE				
305-309—Printing and Binding	43,835.79	37,500.00	6,335.79	
315-316—Advertising and Sales Promotion	21,866.64	22,000.00		133.36
320-322—Copy Sales Promotion	13,103.12	12,000.00	1,103.12	
325—GUIDE Committee Travel	1,705.45	1,400.00	305.45	
	125,606.93	110,600.00	16,136.70	1,129.77
HEADQUARTERS—SCHEDULE I				
210—Salaries	$ 72,175.51	$ 79,200.00	$	$ 7,024.49
210A—Social Security Tax	897.85	1,000.00		102.15
211—Public Relations	4,842.00	5,500.00		658.00
212—Travel—Secretary and Staff	2,581.06	2,600.00		18.94
213—Rent and Light	11,254.98	11,200.00	54.98	
214—Telephone	2,008.66	2,000.00	8.66	

HEADQUARTERS (*Continued*)				
215—Telegraph	615.12	500.00	115.12	
216—Postage	5,315.86	5,000.00	315.86	
217—Printing and Stationery	6,622.69	6,000.00	622.69	
219—Addressing and Address Changes	366.27	500.00		133.73
220—Professional Services	3,363.98	3,000.00	363.98	
221—Bank Charges	49.12	100.00		50.88
222—Insurance	341.22	1,000.00		658.78
11—Furniture and Fixtures	2,286.02	2,000.00	286.02	
223—Office Expense (Miscellaneous)	2,796.67	2,000.00	796.67	
224—Pension	1,176.28	1,000.00	176.28	
	116,693.29	122,600.00	2,740.26	8,646.97
RESEARCH				
Research Committee Expenses	4,269.59	5,000.00		730.41
Staff Salaries	80,771.59	87,170.00		6,398.41
Laboratory Expenses	22,966.81	24,600.00		1,633.19
Laboratory Operation and Maintenance	12,747.90	10,300.00	2,447.90	
Provision for Building Maintenance	2,000.00	2,000.00		
Research Contracts and Fellowships	28,454.52	27,000.00	1,454.52	
Navy Research	17,839.83	24,840.00		7,000.17
	169,050.24	180,910.00	3,902.42	15,762.18
FUND RAISING—SCHEDULE K				
Salaries and Fees	12,012.93	11,700.00	312.93	
Travel	594.75	1,000.00		405.25
Printing	897.58	2,500.00		1,602.42
Postage and Miscellaneous	229.63	1,200.00		970.37
	13,734.89	16,400.00	312.93	2,978.04
TOTAL DISBURSEMENTS	$451,227.55	$462,160.00	$ 24,347.58	$ 35,280.03

The chairman appointed the following tellers of election: W. G. Boales and W. H. Old, both of Detroit, Mich., and G. V. Zintel, Chicago, Ill.

Following the presentation of two technical papers (see Program, p. 31) Reg. F. Taylor, chairman of the First Session, introduced M. W. Bishop, president of Illinois Chapter, who welcomed the members and guests to the 59th Annual Meeting sponsored by the Illinois Chapter, the oldest chapter in the Society. Mr. Bishop in turn introduced W. A. Kuechenberg, general chairman, Committee on Arrangements, and G. W. Bornquist, vice chairman. He gave credit to both for the work of organizing the various committees and making arrangements for the Annual Meeting. Second Vice Pres. L. N. Hunter responded to Mr. Bishop's welcome, and thanked the Chapter in the name of the Society.

Mr. Kuechenberg introduced Dr. Preston Bradley of Chicago, who gave an inspirational talk on the subject, *Today—Tomorrow and You.* Chairman Taylor

thanked the authors of the technical papers and expressed appreciation to Dr. Bradley, and then adjourned the meeting at 11:45 a.m.

SECOND SESSION, TUESDAY, JANUARY 27, 9:30 A.M.

Treas. J. Donald Kroeker called the second session to order at 9:30 a.m. in the Grand Ballroom of the Conrad Hilton Hotel.

W. G. Boales presented the Report of the Tellers of Election as follows:

REPORT OF TELLERS OF ELECTION

Ballot for Officers	Total
President, Reg. F. Taylor, Houston, Tex.	1853
First Vice President, L. N. Hunter, Johnstown, Pa.	1853
Second Vice President, John E. Haines, Minneapolis, Minn.	1840
Treasurer, John W. James, Chicago, Ill.	1853
Members of Council (Three-year Term)	
I. W. Cotton, Indianapolis, Ind.	1851
A. W. Edwards, Cincinnati, Ohio	1852
L. T. Mart, Kansas City, Mo.	1852
E. R. Queer, State College, Pa.	1852
Member of Council (One-year Term)	
Bruce L. Evans, St. Louis, Mo.	1852
Committee on Research (Three-year Term)	
R. S. Dill, Washington, D. C.	1850
F. H. Faust, Bloomfield, N. J.	1852
A. J. Hess, Los Angeles, Calif.	1851
H. A. Lockhart, Morton Grove, Ill.	1852
J. W. McElgin, Philadelphia, Pa.	1852
TOTAL BALLOTS RECEIVED	1858
TOTAL LEGAL BALLOTS	1853
INVALID BALLOTS	5

Scattered votes: 2nd Vice President 13; Members of Council 5; Committee on Research 8.

Respectfully submitted,

W. G. BOALES, *Chairman*
W. H. OLD
G. V. ZINTEL

Chairman Kroeker announced that the new officers, members of the Council and members of the Committee on Research would be installed at the final session on Thursday, January 29. The four technical papers were presented and discussed following the announcements by Chairman Kroeker.

THIRD SESSION, TUESDAY, JANUARY 27, 2:00 P.M.

Second Vice Pres. L. N. Hunter called the third session to order at 2:00 p.m. and announced that the session would be a symposium on Research Administration.

He pointed out that research in industry and in the heating, ventilating, and air conditioning profession is becoming of greater importance as time goes on. It is apparent that the future of the profession is dependent upon the success of research and the amount of research that is carried on. The increasing popu-

larity of research poses a number of problems, especially those of organization and administration. The administration of research had therefore been selected as the theme for the discussion to follow.

Dr. E. A. Walker, State College, Pa., spoke on *Participation by the Colleges in Engineering Research.*

Dr. Walker said that the growth of research in the U. S. has been very marked during the last ten years with Government sponsored projects increasing from $350 million in 1940 to $1.3 billion in 1952 not including atomic energy projects. The National Science Foundation set up by Congress in 1950 and the Army and Navy have sponsored basic and applied research primarily in Universities and Colleges.

The growth of engineering schools in recent years has resulted in two criticisms of the profession: (1) the engineering curriculum has not included enough studies in the humanities; (2) the engineer is too narrow scientifically.

To combat these criticisms, studies in the social sciences have been added to many of the engineering curricula while other engineering colleges are giving courses in engineering sciences and physics which will give the student a broad background in the physical sciences.

The problem of teaching research techniques to engineers has been solved to some degree by having graduate students work with investigators on various research projects. Research has been defined as *man's search for an understanding of the physical world.* With the growth of research in colleges, the need for efficient organizational structure has become apparent. Many colleges are setting up departments of research or research institutes separate from the teaching divisions. The assignment of good men to problems which are basic in nature and aiding them with small numbers of assistants and graduate students who can profit by the guidance, will provide universities with men trained in the techniques of research. This will enhance the university's position and provide a continuing supply of trained research engineers for government and industry.

R. F. Legget, Ottawa, Ont., Canada, discussed *Government Sponsored Research—A Canadian View.*

He pointed out that the participation of Government in research has been an accepted fact for sometime in Canada and there is a happy integration of research by government, industry and universities. Government research in Canada began in the field of agriculture and today Federal research is under the direction of the National Research Council of Canada set up in 1916. One task of the Council is to increase the number of trained research workers. The Council is served by associate committees with representatives from Government, schools, industry, and labor.

The demands of World War II resulted in a tenfold increase in National Research Council personnel to over 3,000. Funds for the operation of the Council are voted annually by Parliament. The staff operates through seven divisions including building research. This division was set up in 1941 and its primary responsibility is housing research and revision of the National Building Code. Many different problems such as cold weather, permafrost, fire research, foundation and soil problems are studied.

The current budget is about $500,000 and projects are undertaken jointly with industrial corporations and other laboratories. The problem of securing personnel for the division is a vital one as government salaries are not equal to those of industry. However the tenure of the staff members is an indication of the role of the Council in the Canadian community.

The object of Government sponsored research in Canada is the improvement of the utilization of material resources in standards and equipment for the physical basis of the lives of all Canadians. Upon this the well-being and progress of the country ultimately depends.

W. H. Kliever, Minneapolis, Minn., presented the third discussion, *Industrial Research in the Control of Environment.*

He stated that research is the means of industrial progress. At one time, curiosity was the motivating force of scientific discovery. Today, theory and experimental

techniques are combined to improve research and development. Industrial research provides new products and improvement of products to sell at a profit. Good research programs include not only details or improvements of individual devices, but a study of complete systems.

The budget for research within a company can be determined by selecting the worthwhile projects and estimating their cost. Any research effort should be built on a sound appreciation of the importance of basic research, the research performed without thought of practical ends. It results in general knowledge and an understanding of nature and its laws. The function of applied research is to provide answers to specific problems.

The scientist doing basic research may not be interested in the practical application of his work, yet the progress of industrial development would stagnate if basic research were long neglected.

Researchers must be capable of analytical approach and yet be original in their thinking. They need freedom of thought and appreciation of their efforts. Research should analyze what the company is doing—to question it—to suggest improvements in products—or to suggest new products. Effective research administration demands good personnel with natural motivation, completely informed of the company problems, and suitable facilities, with a research atmosphere of freedom.

Chairman Hunter thanked the speakers and those who had made comments, and adjourned the meeting at 4:30 p.m.

FOURTH SESSION, WEDNESDAY, JANUARY 28TH, 9:30 A.M.

Second Vice Pres. L. N. Hunter called the fourth session to order at 9:30 a.m., and reported that the registration at that time had exceeded 2500 members and guests. Three technical papers were then presented and discussed in considerable detail.

FIFTH SESSION, WEDNESDAY, JANUARY 28TH, 1:30 P.M.

First Vice Pres. Reg. F. Taylor called the fifth session to order at 1:30 p.m. in the Grand Ballroom of the Conrad Hilton Hotel. He introduced I. W. Cotton, chairman of the Committee on Research, who presented the 1952 Annual Report of the Committee on Research.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1952

DURING the operating year 1952 Society research was concerned with 13 active projects at the Laboratory and 14 projects at 12 cooperating institutions. The Report of the Director of Research which forms a part of this report gives details of these activities.

There were three meetings of the Committee on Research, three meetings of the Research Executive Committee and a total of 24 meetings of Technical Advisory Committees—with a total attendance of over 240 Committee members and over 200 guests.

Papers prepared for publication and accepted numbered 13. During the year the Laboratory staff organized its own screening and editing system in an effort to make papers and progress reports readily readable.

The Director recognizes in his portion of the report the fact that papers do not constitute the end objective of research—rather, they represent the first step in the real job of getting information to our membership and the public in definitely useable form. Continued effort will improve the presentations.

During the year the odor-test rooms have been substantially completed. This structure joins the Environment Laboratory on the main floor of the Research Laboratory as an important capital addition, with a decided difference, however, in objectives and scope.

The Environment Laboratory has already proved its practicability in the assembly and tabulation of much needed data regarding heat exchange between panels and the enclosed space, a type of research which produces new knowledge for use in present practice in panel heating and cooling.

Odor studies fall into an entirely different category. No clear definition of odor, either physical or physiological, has yet appeared; in this project the Laboratory is tackling a problem which is known to exist, but cannot be accurately described. Such research represents not only a challenge but also the courageous expenditure of money by the Society.

Further efforts have been made throughout the year to cooperate more effectively with other Society committees—Guide, Standards, Program and Papers, and Publication. Information about matters of general interest in connection with Society Research has been made available to our membership, both directly by correspondence and at Chapter meetings both as complete programs and as short reports. The Committee on Research, as individual members and as a Committee working through the Laboratory, has established and is maintaining effective liaison and cooperation with a great number of other laboratories and investigators in our fields of interest in many parts of the world. As reports from abroad come to the Research Laboratory the information they contain is distributed through the Committee and the Technical Advisory Committees.

Progress reports to the Technical Advisory Committees and to the Committee on Research by the Director of Research have attained a new high in promptness, accuracy and completeness. This year a comprehensive report to the Council carrying the cover title, A.S.H.V.E. Research—1952 Accomplishments and 1953 Programs, was submitted in time for study by the members prior to the final meeting of the 1952 Council.

In 1950 a survey was made to determine membership interest in the existing and potential activities open to our research. The Society's long-range research program, as requested by Council, was then prepared on the basis of the results of this survey which had shown a gratifying response from our members. This year another survey has been implemented by direct effective cooperation of the Society's Executive Secretary. A long list of items of present and future potential interest for Society research was sent to the membership along with annual dues statements over a period of some weeks. Each member was requested to indicate his individual suggestions by marking all items of particular interest to him, and with spaces for his written additions and comments. Once again membership interest has been indicated and this survey will prove a valuable guide to the Committee on Research in its continuous responsibility of keeping the research program, both present and long range, representative of the usefulness desired by the membership which makes it possible.

The close of the present year coincides with the close of a full decade of change and progress in Society research. In 1943 the Society moved its research facilities from quarters located for many years in the Bureau of Mines at Pittsburgh. Space was rented in Cleveland; the present Director undertook the recruiting, training and assigning of a totally new staff in new quarters. Shortly afterward larger space was acquired outright by the Society for its research and another move into commodious and well arranged space had to be made. During all this period the research program was being carried forward and developed into an expanded program. A relatively short time ago a change in operating methods of the Committee on Research, the Technical Advisory Committees and to some extent the staff, became necessary because of the adoption of the Society's new By-Laws. Comprehensive Operational Guides were carefully written, considered and adopted by the Committee for submission to Council, and by Council authorization were substantially instituted. Concurrently, the directive of Council to extend cooperative efforts with other institutions was followed, and important projects set up for cooperative research.

All of the activities mentioned were superimposed on the daily requirements of an

increasingly dynamic research program. Research achievement by the Society, during the decade just completed, has not been permitted to suffer, which makes the record of solid accomplishment truly remarkable. Now that all of this extra work of change and expansion has been completed, Society research faces a period of concentration on the program itself, and can confidently be expected to do the kind of work expected of it by a membership devoted to the advancement of our science and arts.

There were some marked increases last year in the number of industrial organizations contributing general funds for Society research, and, as a consequence, an increase in the total amount contributed.

During the Semi-Annual Meeting certain Society Officers suggested that no matter what benefit the industry might derive from its own developments and activities, the electrical load increases with practically every advance which the industry makes, thus, directly benefitting the electric utility industry. In view of this, it was suggested that an approach be made to this industry to see the extent to which individual organizations in it might be directly interested in support of Society research.

Under the authority of the Ways and Means Committee, and the Committee on Research, the chairman called on a number of electric utilities in the middlewest to discuss Society research and to see whether these organizations felt that they and the Society had a community of interest. These visits were not concerned with direct requests for financial support, but merely to find the grounds, if any, of mutual interest.

During recent weeks the chairman called on nine such utilities and discussed generally with a group of the top officials of each of these companies, some of the problems of the utilities and what A.S.H.V.E. research might be able to offer in helping to solve some of the problems connected with the load for which the industry might be considered responsible.

Not in a single instance was he told that there was no community of interest. Some of the companies showed considerable interest and immediately took steps to insure a closer liaison between them and the Society by suggesting that individual members of their staffs apply for Society membership. Two of the companies have already given active support to the research program. It is believed that this example will be followed by others and that the utilities, as a whole, might prove to be very acceptable supporters of Society research in the future.

The committee has taken steps whereby officers, past officers, members of the Council and members of the Committee on Research can assist in telling the story of A.S.H.V.E. research to the utilities in the particular location where they do business or where they live. A special brochure has been prepared to assist in these visits. It may interest the Society membership to know that in the more than 60 cities where the Society has strong representation of membership, there are about 26 million electric meters installed.

The Committee on Research fully realizes what such an expansion of the base of its general contribution means in the way of added responsibilities. The committee now is required to make good to our own membership and to those who have supported and made possible the expansion of Society research in recent years. Any broadening of the Society's base of general contributions will increase its responsibility to those who are supporting the research work.

The Committee on Research recently received a letter from a member of the official family of this Society. He said that for some time he had felt that as one of those who made a livelihood from the industry, he owed something toward its continued progress. He thought that the best way to meet this self imposed obligation was to support Society research financially because with a sound research program there would be no question about the future industry of which he was a part. His very constructive and helpful letter and the check enclosed with it, were a source of considerable encouragement to the Committee on Research. The committee has been told by these men that they intend to see from an examination of their earnings, what contribution

they might make to Society research as an individual business investment toward a continuing expansion of our industry.

"With the presentation of this report, my term as Chairman of the Committee on Research comes to an end. May I say that I have enjoyed every minute of these last two years. The chairmanship of the Committee on Research has become a big job, with tremendous opportunities. I have enjoyed a great measure of cooperation which I appreciate to the full and for which I now wish to express my thanks to all members of a magnificent Society. In saying thank you and goodbye as Chairman of the Committee on Research, I want, if I may, to leave one final thought with the members of this Society; that thought is *I believe in Society Research.*"

Report of the Director of Research

Cyril Tasker, director of Research, presented his report in which he outlined some of the details of the 1952 research program.

1952 was a busy year; there were many times when additional graduate engineers could have been used very effectively. We were somewhat disappointed that so few of the June graduates in 1952 seemed interested in A.S.H.V.E. research fellowships at the Laboratory, for we feel that this is a worth-while plan.

Dr. H. B. Nottage resigned from the staff in August to undertake important research and development work in Southern California. His unusually well-developed critical and analytical qualities will be missed at the Laboratory where he contributed much of value to the Laboratory's work in many fields over the past seven years. His outstanding contributions to fundamental research in air distribution have already been recognized, not only by fellow experimentalists and research investigators, but by some of the leading *practical engineers* in the industry. Fortunately, he will be able to continue to assist the Society in some of its research program by serving on Technical Advisory Committees.

The Committee on Research recognized, in September, the fine services which Miss I. M. Jahn has rendered in research operations over the past eight and one-half years by approving her appointment as Administrative Assistant to the Director of Research.

Equipment and Facilities

As in the previous year, 1952 saw important additions made to the Laboratory's inventory of scientific apparatus and equipment. The Odor Test Rooms, costing about $25,000 to design, build and equip, were the largest single addition. We are certain that many uses will be found for these facilities even over and above their use in the odor program.

We should look ahead to the time when we can add modern computing apparatus to the Laboratory's inventory. More and more we see the need for this in the solution of complicated heat flow problems, the solution of which by *long-hand* methods becomes impractical. There may also be need for equipment to analyze masses of data such as may be obtained in the odor program or from the Environment Laboratory.

Minor improvements were made to the property, which is being maintained in good condition and has long since justified the decision that the Society took in 1946, when it was purchased.

Guide

By making the Chairman of the Committee on Research an ex-officio member of the Guide Committee, liaison is maintained between the two committees. The research man is often apt to think of a research paper as the end point of his work, but, as the present Chairman has pointed out in some of his talks to the Chapters, this is only the beginning for many of the members of the Society, the end point being the translation of the results of research into practical usable information in The Guide.

Members of several Technical Advisory Committees have participated in the revision and rewriting of Guide chapters, and in many cases pertinent chapters have been reviewed by TAC's as a whole.

This is a healthy kind of cooperation which could well be expanded. More might be done by adding to the Laboratory staff personnel experienced in the art of technical writing who could translate research results into practical data.

AIR FLOW AND DISTRIBUTION

Evaluation of Past Work: During the year a sub-committee of the TAC on Air Distribution, with W. A. Grant as Chairman, made a careful study of all work performed under the Committee, both at the Laboratory and the cooperating institutions, in the past decade. The report provides an extremely valuable summary and analysis and contains important recommendations on future Society research in this field.

Isothermal and Chilled Ventilation Jets (A.S.H.V.E. Laboratory): The doctorate thesis of H. B. Nottage, covering studies on isothermal jets and on a chilled jet, was submitted to Case Institute of Technology early in 1952. A limited number of copies were distributed to colleges, to the Navy, and to industrial organizations especially interested in basic work on air distribution.

The evaluation sub-committee said *This work is most impressive. We predict that as time passes it will be regarded as one of the Society's great contributions to the advancement of applied science.*

Three papers were presented in June 1952 to summarize certain phases of the work and supplement papers presented in January 1952 and earlier.

Air Velocities in Ventilating Jets (Case Institute of Technology): Cooperative work continued during 1952. Prof. G. L. Tuve completed a report (presented at the 59th Annual Meeting) on *Air Velocities in Ventilating Jets* in which he reviewed various methods suggested by other investigators for analyzing jet performance and showed that they supported proposed methods for determining velocity patterns along the jet axis, velocity profiles, and the angle of divergence. Experimental data were included for axial and radial annular jets and for air streams flowing along a wall.

Downward Travel of Heated Jets (Kansas State College): Cooperative work continued during 1952, with assistance from the *Industrial Unit Heater Association.* Prof. Linn Helander completed a report (presented at the 59th Annual Meeting) on the *Maximum Downward Travel of Heated Air from Standard Long Radius A.S.M.E. Nozzles.* Graphs and formulas having practical guidance for air conditioning engineers and designers of unit heaters were included. Studies were also made on one commercial type of down-blow unit heater in an attempt to develop suitable conversion factors for the various types of industrial units.

Friction in Rectangular Take-Off Fittings (Michigan State College): Studies on the resistances of different types of rectangular fittings designed to accomplish divided air streams were under way during 1952. Results given in a progress report show that the resistance of well designed fittings under certain conditions is greater than commonly thought, and that straight-through branch fittings may offer as much resistance as take-off branch fittings under certain conditions. The studies are continuing, to cover a wider range of basic designs.

HEAT FLOW AND HEAT TRANSFER

Cooling Loads: Continuing their studies, made at the suggestion of the TAC on Cooling Loads, Profs. C. O. Mackey and N. R. Gay presented, in June, a paper entitled, *Cooling Load From Sunlit Glass,* in which they developed mathematical solutions relating the instantaneous cooling load per square foot of floor to the instantaneous rate of gain of radiant energy. The mathematical results were compared with the results obtained with a fluid analogue.

The work carried on at the Laboratory under the TAC on Heat Flow Through Glass during 1952 is of particular interest in connection with cooling loads. The two committees have maintained close liaison.

Air Conditioning Thermal Circuit: During 1951 H. B. Nottage prepared a thermal-circuit diagram which was designed to portray and interrelate the major factors which influence the instantaneous equipment load. After careful review and analysis by many interested persons, it was recommended that an exploratory project be undertaken at the Laboratory.

To test the feasibility of this method of approach and to gain additional experience, the equations for a single-room system with northern exposure, one case with glass block walls and the other with ordinary window glass, were set up and solved with the aid of a general-purpose calculating machine (the property of Case Institute of Technology). The results show the differences between the actual cooling load and

the instantaneous heat gain calculated by the methods used by most air conditioning engineers and given in The Guide. A research paper is being prepared to report the results of this initial study.

Lighting Loads: During the year there was a renewal of discussions with the *Illuminating Engineering Society* on the problem of heat gain due to lighting loads. This question is becoming of increasing interest to air conditioning engineers and to the *I.E.S.*

Shading Studies: The first results of Laboratory studies on the shading of sunlit glass were presented in June in a paper entitled *The Shading of Sunlit Glass: An Analysis of the Effect of Uniformly Spaced Flat Opaque Slats.* The report covered an analytical investigation of the transmission, absorption and reflection of solar radiation by slat-type sun shades such as venetian blinds, jalousies, and some types of sun screens.

During the year the experimental investigation was continued on a number of different slat-type sun shades to supplement the data developed analytically. Transmittance and absorptance of solar radiation were determined; shade color, slat-width, slat-spacing ratio, slat angle, and the position of the sun were the principal factors studied. A second research paper on this work entitled *The Shading of Sunlit Glass: An Experimental Study of Slat-Type Sun Shades* was presented at the 59th Annual Meeting.

Heat Pump

Moisture Migration in Granular Media: Working under a cooperative contract, Prof. W. A. Hadley and Raymond Eisenstadt of Columbia University completed a study of the transfer of moisture in a granular medium due to a temperature gradient. Using a radioactive salt and special techniques, including a Geiger counter, they were able to show that moisture may move in such a medium either by capillarity or by vapor diffusion. The results indicated that there is a critical moisture content which determines the mode of moisture movement.

Since moisture plays an exceedingly important role in the dissipation or absorption of heat wherever an installation involves hot or cold pipes or cables, buried underground, the results are of significance in the consideration of heat pump ground coils. A paper has been prepared on this work; it is hoped that it will be published in the spring of 1953.

Solar Energy as a Heat Source: Studies which have been under way since early in 1950, under a cooperative research contract with the University of Minnesota, are concerned with systems by which solar energy may be utilized either as a partial or as a total source of heat for operation of heat pump heating systems. Since the availability of solar energy was one of the prime considerations, studies have been completed on the radiation available on horizontal, south-vertical and south-tilted surfaces on cloudless days.

The relative heating demands for both design conditions and for total yearly heating load are being studied, together with the ratio of design heating load to design cooling load conditions.

Components of heat pump systems for utilizing solar energy are being studied, with particular attention to collectors, their types and locations, their efficiencies, both instantaneous and daily, as well as the optimum temperature of collection. Storage systems are being reviewed as to the types of materials, the temperature of storage and the location and types of storage bins.

Heating Loads—Infiltration: Using equipment purchased by the Society and loaned to the National Bureau of Standards, the Heating and Air Conditioning Section of the Bureau made a study of infiltration in one room of the N.B.S. Test Bungalow, using the tracer-gas technique. During this work the Bureau also developed another instrument for measuring the rate of decay of tracer-gas and made comparisons of the results obtained with the two instruments.

Although the results to date are not considered to be wholly conclusive, interesting data were secured on the effect of an open fireplace flue. Tests showed that, under similar outdoor and indoor conditions, the air change rate increased from less than 1 per hr with a closed flue to about 2 per hr with an open flue.

HOT WATER AND STEAM HEATING

Noise in Piping: The studies at Northwestern University, being made under a cooperative research contract, were designed to provide information on the generation and transmission of noise in water piping systems as they are related to velocity, water temperature, pressure, pipe size, design configuration and pump vibration, and to discover methods by which noise levels can be reduced. They have been under way throughout 1952.

Since the vibrations (sounds) produced by the piping components could appear either as fluid-borne or as structural-borne vibration, both aspects of the sound transmission were of interest. Provision was made for measuring air-borne sound by the use of a microphone inside a live sound chamber enclosing the test pipe. Vibration pick-ups have been used to detect pipe-wall vibration, and attempts have been made to measure the sound level and frequencies remaining in the water by means of a microphone mounted through the pipe wall into the flowing stream.

Although, as yet, only qualitative information has been obtained, the investigators have concluded: (1) that cavitation is an important factor, (2) that straight runs of pipe do not appear to contribute significantly to the production of noise, at least up to velocities of 10 fps, (3) that noise generated at valves and fittings far exceeds that for straight pipe runs, (4) that while contractions of flow take place quietly, sudden expansions of the flow stream, and also sharp directional changes, produce noise.

Water in its Metastable State in Heating Boilers as a Possible Explosion Hazard: In mid-year the Society entered into a cooperative research contract with Northwestern University to make a study of water in its metastable state in heating boilers as a possible explosion hazard. Water exists in the metastable state when its temperature is above the normal boiling temperature for the pressure existing. A disturbing influence (vibration, surface roughness, etc.) can suddenly cause the water to boil rapidly (with bumping) and give a very rapid increase in volume caused by the sudden flashing. This rapid volume increase could give momentary high pressures which might constitute an explosion hazard.

Air Entrainment in Hot Water Heating Systems: The cooperative research agreement with the University of Florida (Gainesville), to cover the study of the effects of air or gases on the operation of various types of hot water heating systems and methods of eliminating these from the systems, was renewed as of April 1, 1952.

Since results from earlier studies on a simple system were not conclusive, a second system was designed and is being set up with a boiler in the basement and either cast iron radiation or non-ferrous convectors in the basement and on the first or second floors. The piping system has been so designed that by proper valving it may be operated either as a one-pipe system or as a two-pipe reverse return system.

Compression Tank Sizing: Under the sponsorship of the TAC on Hot Water and Steam Heating, H. A. Lockhart and G. F. Carlson presented a report on compression tank sizing at the 59th Annual Meeting. The work covered in the report was carried out with the purpose of presenting a simple, safe and practical method for sizing compression tanks.

PANEL HEATING AND COOLING

Plaster Panel Tests: When the Laboratory paper on plaster panels was presented in June 1951, questions were raised as to whether or not the test panels were representative of the type and quality of plaster panels encountered in the field. In an attempt to answer this question and provide additional information, two additional plaster panels were constructed at the Laboratory. Both had non-ferrous tubes located 4 in. on center above metal lath with the lath wired to the tubes at 8 in. intervals. Three coat plaster jobs were applied to each panel. On one panel, special care was used to get unusually good tube embedment, while on the other the tube embedment is poor and yet probably not unrepresentative of many panels as built in the field. The results obtained with the two panels should indicate differences in performance which might result from variations in plastering. Tests on one panel were completed in 1952; tests on the second panel were under way at the year's end.

Field Tests: Additional field test data, collected in four test houses during the 1950-51 heating season, have been analyzed. In general, the results check those obtained from the previous year's tests, the results of which were published in January 1951.

Concrete Floor Slabs: In 1950 the Laboratory presented a paper on the heat-flow analysis for a concrete floor slab having tubes located within the slab and having both surfaces isothermal. Later, a study was made on a second case, in which the tubes are located at the slab-earth interface with the bottom of the tubes tangent to the interface. A practical example of this would be a floor slab laid directly on the earth with pipes at the bottom of the slab.

Although a general mathematical solution was developed to cover this case, most of the application data were developed by means of an electrolytic analogue because of the complexity of the mathematical treatment. A paper covering this work has been recommended for publication.

Heat Exchanges in a Ceiling Panel Heated Room: The Environment Laboratory, put into actual operation in November 1951, was in daily use throughout 1952. Whenever sufficient staff were available, the room was operated on a two-shift per day basis.

The initial work covered an investigation of heat exchanges in a ceiling panel heated room. The variables investigated were (a) panel surface temperature, (b) floor and wall surface temperature, (c) infiltration air quantity, and (d) infiltration air temperature.

Results from these first series of tests are included in the paper, *Heat Exchanges in a Ceiling Panel Heated Room* (presented at the 59th Annual Meeting).

Floor Panel: When these tests were completed, another series was started in which heat was introduced through the floor instead of the ceiling. The general scope of the floor panel tests and the variables involved were the same as for the ceiling panel tests. This series was substantially completed by the end of the year.

Control of Panel Heating Systems: Under a Research Fellowship established at the University of Minnesota (under Prof. A. B. Algren) field studies were made in the 1951-52 heating season in Minneapolis and Milwaukee to develop information on the optimum method of control for each basic type of heating panel installed in various types of structures with varying percentages of glass area.

The 1952 studies were made in (1) a single-story general office building in Milwaukee with a floor panel system (2) the factory area of a plant in Milwaukee, with six zones of floor panel, four in the manufacturing space and two in the office section (3) a residence in Minneapolis, floor area of about 1450 sq ft, with a floor panel heating system consisting of five continuous coil circuits, the residence being divided into four zones.

Physiological Research

The College of Medicine, University of Illinois, working under a cooperative research contract supported jointly by the U. S. Public Health Service and the Society, completed in 1952 the reports of two studies and submitted them for publication. One paper was presented at the 59th Annual Meeting, the other is tentatively scheduled for the Semi-Annual Meeting in 1953.

Physiological Adjustments of Clothed Men and Women: The first paper reports on the physiological changes in healthy young women, clothed in union suits or in summer clothing, on entering a hot moist room and on leaving it. Similar data on young men were reported in January 1952. The experiments were designed to impose stress on the processes concerned in the adjustment of the subjects to heat exposure much as does leaving an air-conditioned space for the outside of a hot humid day and returning to the cooled space with clothing saturated with sweat. The studies showed that there was no significant difference between men and women in their ability to adjust to sudden changes in environment. Since both sexes adjust with equal promptness, the presence of men and women in the same space does not require alterations in accepted specifications for its air conditioning.

Effect of Relative Humidity on Heat Loss of Men Exposed to Different Environments: The second report covers the results of studies on five fasting men, either nude or dressed in union suits, and exposed to environments of different relative humidities and the same dry bulb temperature; it also covers studies on ten young men and eight young women exposed to environments with widely differing humidities.

The results showed that effects attributed to relative humidity appeared more rapidly at higher environmental temperatures.

O.N.R. CONTRACT

Human Calorimetry: The calorimeter which has been designed and constructed at the Laboratory, under a contract with the Office of Naval Research, for the study of heat exchange between the human body and its surroundings, was substantially completed, according to the original design, when the contract period ended on June 30, 1952. At that time the contract was renewed to September 30, 1953, so that additional facilities could be incorporated to permit very accurate control of the air to be used for respiration by the subject during a test.

The equipment required for these additions to the calorimeter is even more complicated and delicate than that developed and assembled to date. Though its design, construction and operation presented a number of new problems, good progress had been made by the end of the year in solving most of them and in assembling the equipment.

The originator of the idea behind this calorimeter, Dr. T. H. Benzinger, and his co-worker, Dr. Charlotte Kitzinger, spent several weeks at the Laboratory in November and December making test runs on the calorimeter and on the equipment for the ventilation air circuit. Although, as was expected, the test runs revealed a number of minor points on which further work might be beneficial, the equipment on a whole operated well up to expectations.

Low-Inertia, Low Resistance Heat Flow Meter: A research paper on the construction and use of a special heat flow meter, developed during the O.N.R. contract, was presented at the Semi-Annual Meeting in June. Since that time, the Laboratory has supplied even more detailed information on both the materials and the methods used in the construction of different sizes and types of heat flow meters. A number of organizations have sent representatives to the Laboratory to see how these meters are made and used.

ODORS

Special Test Rooms: The construction of the odor test rooms and their various auxiliaries was virtually completed during the 1952 fiscal year. All of the interior surfaces of the rooms and corridor are made of a porcelain enameled wall board, selected to provide surfaces which would have low odor absorption and could be easily cleaned.

A separate air conditioning system was provided for each of the test rooms and the corridor, thereby making it possible to independently control the temperature and humidity in each space. The conditioning equipment is located above the rooms. The isometric drawing shows the system for room No. 1; the system for room No. 2 is identical. In addition to controlling the temperature, humidity, and cleanliness of the air entering the rooms, these systems provide for the measurement of air flow, and for the introduction of odors into the air streams. The system for the corridor is identical with those for the rooms, except that in the corridor flow measurement and odor introduction are not required.

Program: An odor test program has been drawn up after consultation with Technical Advisory Committee members and others. Since the first step will be the training of an odor panel, members of which will be selected from the present Laboratory staff, visits were paid, during December, to other laboratories which have specialized in odor work, to discuss problems of panel training and testing techniques with the men in charge.

The first actual testing is being planned to try to develop information regarding the effect of temperature and relative humidity on odor perception.

PLANT AND ANIMAL HUSBANDRY

In mid-year, the Chairman, A. J. Hess, completed his organization of the Committee which has six engineering members and seven members eminent in the fields of plant physiology, botany, dairy husbandry, animal nutrition and biology. As part of its activities, this Committee, which is quite unique among Society Technical Advisory Committees, will endeavor to translate the results from past and current research in plant physiology and animal husbandry into information useful both to those who desire to have optimal environmental conditions for animals and for growing plants,

and for the engineer who is called upon to design systems to maintain such optimal conditions.

Role of Air Conditions in Plant Growth: Studies have been under way during 1952 at the Earhart Plant Research Laboratory of the California Institute of Technology, Pasadena, under a cooperative research agreement with the Society. The basic purpose of this work was to study the role of air conditions in plant growth, plant reproduction and photosynthesis, the particular objective being the establishment of the optimal greenhouse conditions for the growing and fruiting of tomatoes.

Tests made to determine the variation in optimal growing and fruiting conditions for different varieties of tomatoes yielded very interesting results, which have clearly indicated some of the possibilities that exist for air conditioning of tomato greenhouses. Experiments made to determine the effect of the rate of air movement past the plants showed that the best plants were produced when the rates were relatively high (100 to 200 fpm).

The experimental program as originally defined was substantially completed by the end of the year and a complete report was in preparation.

AIR CLEANING

Filter Test Methods: The joint efforts of the Laboratory and the members of the Technical Advisory Committee resulted in the development late in 1951 of the third draft of proposed testing procedures for rating air cleaning devices used in general ventilation. In mid-year, this draft, with detailed drawings included, was distributed for further study.

Shortly afterward the Society was advised that the *Air Filter Institute,* which is composed of twelve manufacturers of air cleaning devices, planned to develop proposed test methods for mechanical-type filters and that Society cooperation would be welcomed.

Test Dusts: General agreement is still lacking on the question of the most suitable test dust or dusts for use in filter testing, the question on which previous efforts to secure a suitable code have foundered. The *A.F.I.* proposed code uses fine Arizona road dust, carbon, and lint. Although efforts by the Laboratory staff early in 1952 to secure sized fractions of spherical glass beads for possible use as a test dust met with very limited success, these efforts are being continued with the cooperation of certain industrial organizations.

THE INDUSTRIAL ENVIRONMENT

During the year the Technical Advisory Committee was reorganized into 6 groups covering respectively the following fields: Supply Air Conditioning; Process Aerodynamics; Air Contaminants; Temperature Extremes; Exhaust Air Cleaning (Air Pollution Control); Industrial Exhaust Systems.

Design of Local Exhaust Ventilation for Hot Industrial Processes: Under a cooperative research contract with the Society, the Graduate School of Public Health, University of Pittsburgh, has been making a study of the basic principles of local exhaust ventilation for hot industrial processes. The ultimate purpose of this study is to determine the exhaust ventilating rate required to capture contaminated air rising from hot processes so that design data can be developed.

Experimental *hot bodies,* capable of giving surface temperatures of from 650 F to 900 F, were constructed at the University and an investigation made of various means for determining the surface temperature of the hot body, and the temperatures and velocities of the convection air currents around the body.

The results of preliminary studies on the nature and direction of convection flow over a hot body seem to indicate that exhaust hoods located not too far above hot operations need not cover the entire hot surface but could be reduced to one-half or three-quarters of the diameter of the hot body, depending on the surface temperature.

Spot Cooling: The group covering *Temperature Extremes* has been making a study of available information on spot cooling, by a review of pertinent literature, inquiries about installations at various industrial locations, and inquiries regarding the use and physiological aspects of spot cooling. From an analysis of the information gathered to date it is suggested that though spot cooling is far more widely used than originally believed, it is often misunderstood and frequently misapplied. Design

information and application data are meager and information is greatly needed as to the conditions resulting from undesirable drafts. Experimental work has been suggested.

INSULATION

Thermal Properties of Building and Insulating Materials: The TAC continued, during the year, to study available data on insulating materials and building constructions and recommended changes and additions to the data published in THE GUIDE.

After a two-day meeting in April the Committee submitted a proposal covering the complete review and analysis of all available data, to be made by a full time member of the Laboratory staff. This proposal was approved by the Committee on Research in June, and on October 1, M. L. Erickson, a graduate of the University of Minnesota and a member of the staff of its Engineering Experiment Station, joined the Laboratory staff to undertake this study.

The objectives of the long-range program are (1) to revise and improve Guide data, (2) to develop improved methods for presenting such data, and (3) to study the effect of factors affecting heat flow through building and insulating materials.

As an initial step in compiling the desired information, technical data on materials that are used today as building materials, insulation, vapor barriers, etc., were requested in a personal letter sent to about 400 organizations. The response to this has been encouraging.

Measurement of Moisture Permeation: Under a cooperative research contract with the Society, Prof. F. A. Joy and A. W. Sherdon of The Pennsylvania State College made a study of rapid methods for the measurement of water vapor permeance of thin materials of the type often used as vapor barriers. The studies resulted in the development of a new instrument, an *Orifice Type Permeometer*, on which a patent disclosure has been filed. A paper covering the development and use of the instrument, submitted in October, will probably be published in 1953.

SORPTION

Methods for Testing Sorption Type Dehumidifiers: The Technical Advisory Committee continued its efforts to develop a tentative code covering laboratory methods of procedure for determining the performance characteristics of sorption type dehumidification equipment. A draft of a code was developed in the spring and was under study by the Technical Advisory Committee members, preparatory to its submission through the Committee on Research to the Standards Committee.

Dynamic Characteristics of a Solid Adsorbent: The Technical Advisory Committee favorably recommended to the Committee on Research a proposal submitted by The Pennsylvania State College, the purpose of which was to determine the characteristics, both adsorptive and desorptive, of a solid desiccant (silica gel) under dynamic air conditions. The project was approved by the Committee on Research for activation in 1953.

SOUND AND VIBRATION CONTROL

The scope of the Committee's activities was extended in mid-year to include the evaluation of all factors involved in the measurement and control of sound and vibration in heating, ventilating and air conditioning systems and equipment.

Following discussions on some of the major problems still facing the industry, the Committee recommended that studies be made and specifications developed for the measurement of noise characteristics of centrifugal and axial fans coupled to a duct system.

There is general agreement that a research program in sound and vibration control should be divided into three broad categories to cover (1) problems of interest to the manufacturers of equipment (2) the development of data likely to be of value to design engineers (3) studies primarily concerned with the standpoint of the user of equipment and organized to determine what the user might want and how best to secure it.

COMBUSTION

Pulsation in Oil Flames: In presenting its long range program to the Committee on Research, the Technical Advisory Committee on Combustion indicated that problems

associated with pulsations in oil flames were of major importance and merited a high priority in future research under the Committee. Resonance in gas flames was recognized as a somewhat parallel problem; in fact it was felt that the two problems were basically the same but that the oil problem was, at that time, more acute.

During the year the Chairman and other Committee members visited a number of colleges and institutions at which work has been done on oil flames or which had staff interested in the possibility of research on this subject. Although no definite plans had been formulated by the year end, much useful information had been collected and the likelihood of being able to develop a project became a distinct possibility.

Following Mr. Cotton's Report of the Committee on Research Chairman Taylor referred to his services to the Committee on Research during the past two years, and commended Mr. Cotton on the results achieved. He felt certain that Mr. Cotton would still be rendering service to the Society on various committees during the coming years. He then called upon the authors of the three technical papers, and each in turn presented his paper.

SIXTH SESSION, THURSDAY, JANUARY 29TH, 9:30 A.M.

Second Vice Pres. L. N. Hunter called the sixth session to order at 9:30 a.m. on January 29, and in turn introduced the authors of the three technical papers which were presented.

INSTALLATION OF OFFICERS

At the technical session's conclusion, Chairman Hunter requested S. H. Downs, Kalamazoo, Mich., Society past president, to conduct the installation of the Society's new officers.

Mr. Downs commended outgoing president, Ernest Szekely, for his excellent administration, despite the handicap of illness; then he introduced the new president, Reg. F. Taylor, Houston, Tex. Mr. Taylor expressed cognizance of his responsibilities and his confidence that the Society would continue to grow in stature.

Next to be introduced were L. N. Hunter, Johnstown, Pa., First Vice President; John E. Haines, Minneapolis, Minn., Second Vice President, and John W. James, Chicago, Ill., Treasurer, each of whom acknowledged the honor bestowed.

With appropriate comments, Mr. Downs then presented Council members I. W. Cotton, Indianapolis, Ind.; A. W. Edwards, Cincinnati, Ohio; L. T. Mart, Kansas City, Mo.; Prof. E. R. Queer, State College, Pa., and B. L. Evans, St. Louis, Mo. Mr. Downs then announced the business of the session concluded and presented the official Society gavel to President Taylor, who called for the report of the Resolutions Committee, which was presented by Prof. B. H. Spurlock, Jr., Boulder, Colo., chairman.

RESOLUTIONS

It is only fitting that we honor our dead and it is proper as we lay away, this, the 59th Annual Meeting of our Society that your Committee propose the following resolutions:

WHEREAS, For the 59th time, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has convened for its Annual Meeting, this time in Chicago, a city of great erudition, culture and hospitality, as guests of Illinois Chapter of the Society; and

WHEREAS, Under the capable direction of our Society's Officers and Staff, the technical sessions have been most successful in that much wisdom and useful scientific data were orally presented and knowledge abundantly gained by the listeners; and

WHEREAS, W. A. Kuechenberg, G. W. Bornquist, and M. W. Bishop, assisted by the entire membership of the Illinois Chapter, by arrangement of social activities have given to ladies and gentlemen alike, delightful bits of fun, feast, Fields (Marshall), and Faust, and

WHEREAS, Dr. Preston Bradley in a most entertaining manner extolled the virtues of Chicago and mankind (with certain reservations); and

WHEREAS, The 11th International Heating, Ventilating and Air Conditioning Exposition reached a new peak in quality as well as size; therefore,

BE IT RESOLVED, That we express our sincere gratitude and thanks:

TO the Illinois Chapter, its Officers and Committees, who engineered and operated a most pleasant and streamlined meeting,

TO the windy city, Chicago, for a week of controlled weather and unstinted hospitality,

TO the Chicago Convention Bureau and Hotels' Management for so competently housing and caring for us,

TO the authors of the technical papers and to those who participated in the discussions,

TO the daily press, the technical press and radio for their assistance in making available to the public our Society's accomplishments in improving the comfort and health of mankind,

TO C. M. Burnam, Jr., for his delightful introduction of our guests and speakers, and to Nathaniel Leverone for his special message on This Freedom of Ours,

TO Chicago's own Everett N. McDonnell, who has devoted so much of his time and energy to furtherance of our Society's aims and who was here recognized in token by the F. Paul Anderson Award,

TO Dr. Bradley, who has inspired us to say,
"Good - By God - we've been to Chicago."

Respectfully submitted,

The Resolutions Committee

B. H. Spurlock, Jr., *Chairman*, Boulder, Colo.
F. H. Faust, Bloomfield, N. J.
A. N. Hoss, Portland, Ore.

C. L. Fleming, Houston, Tex., requested an opportunity to offer the first business for the new year, a tribute to President Taylor in the form of a leather bound book containing letters of regard and reminiscence from friends of the new president. This book contained a large number of autographs and also had provision for insertion of additional letters, autographs, and other items of interest which the president might receive during his year in office.

President Taylor expressed his thanks to his many friends who had presented him with the book.

As there was no further business to be transacted, the 59th Annual Meeting was adjourned at 11:30 a.m.

PROGRAM—59th ANNUAL MEETING

The Conrad Hilton Hotel, Chicago, Illinois

January 26-29, 1953

Saturday—January 24

9:30 a.m. TAC on Insulation, M. W. Keyes, *Chairman (Private Dining Room 19)*

10:00 a.m. Finance Committee, J. W. James, *Chairman (Private Dining Room 20)*

10:00 a.m. Standards Committee, L. N. Hunter, *Chairman (Private Dining Room 5)*

12:30 p.m. Ways and Means Committee, C. E. Price, *Chairman (Private Dining Room 3)*

2:00 p.m. Admission and Advancement Committee, A. W. Edwards, *Chairman (Private Dining Room 12)*

3:00 p.m. Executive Committee, L. E. Seeley, *Chairman (Room 13)*

3:00 p.m. TAC on Air Cleaning, A. B. Algren, *Chairman (Private Dining Room 3)*

Sunday—January 25

10:00 a.m. REGISTRATION *(Normandie Lounge)*

10:00 a.m. Council Meeting *(Private Dining Room 1)*

10:00 a.m. 1952-53 Committee on Research *(Private Dining Room 4)*

3:00 p.m. TAC on Insulation, M. W. Keyes, *Chairman (Private Dining Room 12)*

Monday—January 26

9:00 a.m. REGISTRATION *(Normandie Lounge)*

9:30 a.m. FIRST TECHNICAL SESSION *(Grand Ballroom)*
President Ernest Szekely presiding
Call to Order
Reports of Officers and Council
A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment, by Tohru Inouye, F. K. Hick, M.D., R. W. Keeton, M.D., J. Losch, Chicago, Ill., and Nathaniel Glickman, Miami Beach, Fla., presented by Mr. Inouye
Two Methods of Calculating Friction Heads for Pipe Lines, by F. E. Giesecke, New Braunfels, Tex., presented by C. M. Humphreys, Cleveland, Ohio

11:00 a.m. Greetings by M. W. Bishop, President of Illinois Chapter
Response by Pres. Ernest Szekely
Address: Dr. Preston Bradley, Chicago
Subject: "Today—Tomorrow and You"

11:30 a.m. Ladies Luncheon and Miniature Grand Opera (12 noon to 4:00 p.m.—*Kungsholm Restaurant*)

12:30 p.m. Council Luncheon *(Saddle and Sirloin Club)*

1:30 p.m. Chapters Conference Committee *(Private Dining Room 1)*

2:00 p.m. EXPOSITION—Opening of 11th International Heating, Ventilating and Air Conditioning Exposition *(International Amphitheatre and Chicago Convention Bldg.)* A.S.H.V.E. Booths Nos. 905 and 2-325

2:00 p.m. Ladies' Tea and Drama Program *(North Ballroom)*

Monday—January 26 *(continued)*

2:00 p.m. TAC on the Heat Pump, E. P. Palmatier, *Chairman (Private Dining Room 9)*

2:00 p.m. TAC on Industrial Environment, W. N. Witheridge, *Chairman (Private Dining Room 4)*

2:00 p.m. TAC on Panel Heating and Cooling, P. B. Gordon, *Chairman;* Groups A, B and D *(Private Dining Room 2)*

3:00 p.m. GUIDE Committee, B. H. Jennings, *Chairman (Private Dining Room 3)*

7:00 p.m. Committee on Revision of Code of Minimum Requirements for Comfort Air Conditioning, W. L. Fleisher, *Chairman (Private Dining Room 10)*

9:00 p.m. Informal Party and Get-Together *(Grand Ballroom)*

Tuesday—January 27

9:00 a.m. REGISTRATION *(Normandie Lounge)*

9:30 a.m. SECOND TECHNICAL SESSION *(Grand Ballroom)*
First Vice Pres. Reg. F. Taylor presiding
Report of Tellers of Election
Compression Tank Selection for Hot Water Heating Systems, by H. A. Lockhart and G. F. Carlson, Morton Grove, Ill., presented by Mr. Lockhart
The Bacteria and Odor Control Problem in Occupied Spaces, by R. L. Kuehner, York, Pa., presented by the author
Heat Transmitted to the I-B-R Research Home from the Inside Chimney, by W. S. Harris and R. J. Martin, Urbana, Ill., presented by Professor Harris
Fuel Consumption Analysis for Multi-Family Housing Projects, by Rutcher Skagerberg, and J. E. Phifer, Washington, D. C., presented by Mr. Skagerberg

10:00 a.m. TAC on Air Distribution, G. B. Priester, *Chairman (Private Dining Room 10)*

10:00 a.m. TAC on Heat Flow Through Glass, R. A. Miller, *Chairman (Private Dining Room 9)*

12:00 noon TAC on Hot Water and Steam Heating, J. W. James, *Chairman (Private Dining Room 5)*

1:30 p.m. THIRD TECHNICAL SESSION *(Grand Ballroom)*
Second Vice Pres. L. N. Hunter presiding
RESEARCH ADMINISTRATION SYMPOSIUM
Participation by Colleges in Research, by E. A. Walker, State College, Pa.
Government Sponsored Research—A Canadian View, by R. F. Legget, Ottawa, Ont., Canada
Industrial Research in the Control of Environment, by W. H. Kliever, Minneapolis, Minn.

2:00 p.m. TAC on Combustion, T. H. Smoot, *Chairman (Private Dining Room 3)*

2:00 p.m. TAC on Cooling Load, W. E. Zieber, *Chairman (Private Dining Room 4)*

2:30 p.m. TAC on Sound and Vibration Control, H. A. Lockhart, *Chairman (Private Dining Room 6)*

7:00 p.m. Past Presidents' Dinner *(Private Dining Room 10)*

7:00 p.m. TAC on Heating Load, W. S. Harris, *Chairman (Private Dining Room 3)*

7:30 p.m. Committee on Code for Testing and Rating Heavy Duty Furnaces, E. K. Campbell, *Chairman (Private Dining Room 12)*

Wednesday—January 28

9:00 a.m. REGISTRATION *(Normandie Lounge)*

9:30 a.m. FOURTH TECHNICAL SESSION *(Grand Ballroom)*
Pres. Ernest Szekely presiding
Friction Losses in Round Aluminum Ducts, by F. W. Hutchinson, Berkeley, Calif., presented by the author
Pressure Losses in 4-Inch Diameter Galvanized Metal Duct and Fittings, by H. G. Conn, W. G. Colborne, and W. G. Brown, Kingston, Ont., Canada, presented by Professor Colborne
Room Air Distribution Research for Year 'Round Air Conditioning—Part I, Supply Outlet at One High Side Wall Location, by S. F. Gilman, H. E. Straub, Urbana, Ill., A. E. Hershey, East Pittsburgh, Pa., and R. B. Engdahl, Columbus, Ohio, presented by Professor Gilman

10:00 a.m. TAC on Sorption, G. L. Simpson, *Chairman (Private Dining Room 6)*

11:30 a.m. Ladies Luncheon and Miniature Grand Opera (12 noon to 4:00 p.m.—*Kungsholm Restaurant*)

1:30 p.m. FIFTH TECHNICAL SESSION *(Grand Ballroom)*
First Vice Pres. Reg. F. Taylor presiding
Report of Committee on Research, by I. W. Cotton, *Chairman*
Field Studies of Floor Panel Control Systems, by A. B. Algren, E. F. Snyder, Jr., and J. S. Locke, Minneapolis, Minn., presented by Mr. Snyder
Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys, Cleveland, Ohio, presented by Mr. Schutrum
The Shading of Sunlit Glass—An Experimental Study of Slat-Type Sun Shades, by G. V. Parmelee, W. W. Aubele, and D. J. Vild, Cleveland, Ohio, presented by Mr. Parmelee

2:00 p.m. Ladies Tea and Drama Program *(North Ballroom)*

3:00 p.m. Nominating Committee, M. W. Bishop, *Chairman (Private Dining Room 5)*

7:00 p.m. ANNUAL BANQUET *(Grand Ballroom)*
Toastmaster: C. M. Burnam, Jr., Chicago
Speaker: Nathaniel Leverone, Chicago
Subject: "This Freedom of Ours"
Presentation of Past President's Emblem to Ernest Szekely, Milwaukee
Presentation of F. Paul Anderson Medal to E. N. McDonnell, Chicago, by Pres. Ernest Szekely

Thursday—January 28

9:00 a.m. REGISTRATION *(Normandie Lounge)*

9:30 a.m. SIXTH TECHNICAL SESSION *(Grand Ballroom)*
Second Vice Pres. L. N. Hunter presiding
Maximum Downward Travel of Heated Jets from Standard Long Radius *A.S.M.E.* Nozzles, by Linn Helander, Shee-Mang-Yen and R. E. Crank, Manhattan, Kan., presented by Professor Helander
Air Velocities in Ventilating Jets, by G. L. Tuve, Cleveland, Ohio, presented by the author
Cooling A Small Residence with A Two-Horsepower Mechanical Condensing Unit, by H. T. Gilkey, D. R. Bahnfleth and R. W. Roose, Urbana, Ill., presented by Mr. Gilkey

Thursday—January 28 *(continued)*

Installation of Officers
Resolutions
New Business
Adjournment

12:00 noon 1953-54 Committee on Research (organization meeting) *(Private Dining Room 9)*

1:30 p.m. Council Meeting *(Private Dining Room 10)*

Friday—January 30

12:00 noon EXPOSITION—11th International Heating, Ventilating and Air Conditioning Exposition *(International Amphitheater and Chicago Convention Bldg.)* A.S.H.V.E. Booths Nos. 905 and 2-325. Exhibit closes at 6:00 p.m.

COMMITTEE ON ARRANGEMENTS

W. A. Kuechenberg, *General Chairman*
G. W. Bornquist, *Vice Chairman*

Honorary Chairmen

H. M. Hart S. R. Lewis A. C. Willard

Banquet Committee—H. G. Chapin, *Chairman;* C. M. Burnam, Jr., C. A. Imming, P. J. Marschall, R. T. Murphy, E. E. Sundeen, J. F. Tobin.

Entertainment Committee—J. S. Kearney, *Chairman;* W. H. Brackett, G. B. Coffey, A. L. Crump, W. F. Christmann, J. A. Dell, F. E. Moran, H. B. Ross, Henry Seetch, W. A. Stahl, H. G. Swart.

Finance Committee—G. G. Freyder, *Chairman;* H. E. Anderson, H. J. Couch, T. L. Smith.

Ladies Committee—Mrs. G. W. Bornquist, *Chairman;* Mmes. M. J. Bamond, C. M. Barnes, M. W. Bishop, C. M. Burnam, Jr., H. G. Chapin, J. F. Cummiskey, G. G. Freyder, H. G. Gragg, T. V. Johnson, J. S. Kearney, Herbert Kreisman, W. A. Kuechenberg, P. J. Marschall, J. H. Milliken, W. G. Pennington, C. E. Price, G. V. Zintel.

Plant Inspection Committee—C. M. Barnes, *Chairman;* John Gazlay, C. A. Gustafson, F. W. Kramer.

Publicity Committee—T. V. Johnson, *Chairman;* E. G. Avery, M. J. Bamond, S. F. Rost.

Reception Committee—H. G. Gragg, *Chairman;* Herbert Kreisman, *Co-Chairman;* M. J. Bamond, Thomas Cranage, A. L. Crump, L. P. Lang, G. A. Mansinger, R. D. Misch, O. A. Moran, W. G. Pennington, J. J. Philippi, J. C. Scott, L. H. Streb, F. W. Thompson, W. F. Christmann.

Sessions Committee—J. F. Cummiskey, *Chairman;* Norman Bueter, A. M. Bladen, Thomas Cranage, E. H. Gage, N. A. Gregornik, M. F. Koster, Sydney Miner, J. C. Scott, L. H. Streb, E. E. Sundeen, E. R. Teske, R. W. Van Valzah.

1463

No. 1463

A COMPARISON OF PHYSIOLOGICAL ADJUSTMENTS OF CLOTHED WOMEN AND MEN TO SUDDEN CHANGES IN ENVIRONMENT

By TOHRU INOUYE*, F. K. HICK, M.D.†, R. W. KEETON, M.D.**, J. LOSCH††, AND NATHANIEL GLICKMAN***, CHICAGO, ILL.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and by the United States Public Health Service in cooperation with the University of Illinois, College of Medicine, Chicago, Ill.

AN EARLIER PAPER[1] recorded the physiological changes in young men clothed in union suits or in summer clothing on entering a hot moist room from a comfortable room and on returning to the comfortable room. These healthy subjects adjusted with ease to each new condition. This report is now amplified with the data on healthy young women, Table 1, observed under similar conditions so that the two sexes may be compared. The experiment imposes stress on the processes concerned in the adjustment of the subject to heat exposure much as does leaving an air-conditioned space for the outside on a hot humid day and returning to the cooled space with clothing saturated with sweat.

EXPERIMENTAL ROUTINE

Young healthy women, having measurements shown in Table 1, were observed for 1 hr in a comfortable room maintained at 76 F with either 30 or 80 percent relative humidity (RH), corresponding to 68.8 and 73.4 effective temperature (ET), respectively. Seven women were dressed in union suits rigged with

* Research Assistant, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

† Professor of Medicine, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

** Professor of Medicine, Emeritus, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

†† Research Assistant, Department of Medicine, University of Illinois.

*** Member of A.S.H.V.E.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

[1] Exponent numerals refer to References.

multiple thermocouples[2] and eight in summer clothing. The union suit was of a 12 lb/doz, knitted, 90 percent cotton—10 percent wool fabric, whereas the summer clothing consisted of a short length cotton knitted underwear fitted with thermocouples[1], nylon slip and hose, and a dress of cotton broadcloth. The total weight of this assembly was about 20 oz.

Observations of body weight were made with the subject seated on a balance. Skin and rectal temperatures, pulse, blood pressure, and the subjective feeling of warmth or coolness (thermal sensations) were recorded as earlier described[2].

TABLE 1—PHYSICAL MEASUREMENTS OF SUBJECTS

SUBJECT	GROUP[a]	AGE YEARS	WEIGHT POUNDS	HEIGHT INCHES	BODY AREA SQ FT
E.F.	US	24	129	67	18.2
M.L.	US	22	122	68	17.9
D.M.	US	23	169	71	21.1
J.M.	US	23	120	64	17.0
T.M.	US	22	117	64	16.8
M.O.	US	26	120	67	17.0
P.K.	US, SCl	24	115	60	15.9
C.B.	SCl	20	125	60	16.6
L.B.	SCl	20	114	62	16.3
A.K.	SCl	21	146	67½	19.2
J.L.	SCl	22	101	61½	15.3
V.P.	SCl	24	124	66	17.5
G.T.	SCl	21	140	70	19.4
Z.O.	SCl	26	130	65	18.0
Average	US	23.4	127	65.9	17.7
	SCl	22.3	124	64.0	17.3

[a] US indicates subjects participated in experiments wearing union suits.
SCl indicates subjects participated in experiments wearing summer clothing.

The weight loss, determined from changes in body weight, was used to calculate evaporative heat loss assuming minimal errors in weight as the result of respiratory gaseous exchange.

After 1 hr in the comfortable room the subjects walked into an adjoining hot room, maintained at 98.5 F with a 66 percent RH equivalent to 90.2 ET, and remained there, seated on another balance, for 1 or 2 hr. At the end of the hot room exposure they returned to the comfortable room and were observed for an additional hour.

RESULTS

Comfortable Room

One hour stay in the comfortable room permitted the subjects to stabilize with respect to rectal and skin temperatures, pulse rate, blood pressure, thermal sensation, and evaporative weight loss. The values for these observations on similarly clothed women were the same whether the relative humidity was low (30 percent) or high (80 percent) as shown in Table 2. The mean skin temperature of the women wearing summer clothing was statistically significantly higher than

TABLE 2—OBSERVATIONS ON PHYSIOLOGICAL ADJUSTMENTS OF WOMEN DRESSED IN EITHER UNION SUIT OF SUMMER CLOTHING TO SUDDEN CHANGE FROM A COMFORTABLE ENVIRONMENT OF EITHER 30 OR 80 PERCENT RH TO A HOT HUMID ENVIRONMENT

OBSERVATION	PHYSIOLOGICAL ADJUSTMENTS TO HEAT EXPOSURE			
Type of Clothing	Union Suit		Summer Clothing	
Relative Humidity of Previous Exposure	30	80	30	80
Mean Skin Temperature (F)				
Level before heat exposure	91.55	91.51	92.83[a]	92.53[a]
Level after 10 min heat exposure	96.87	95.96[b]	97.26	96.10[b]
Increase in 10 min	5.32	4.45[b]	4.43[a]	3.57[a b]
Level at 1 hr	97.30	97.03	97.33	97.12
Level at 2 hrs	97.82	97.52	97.51	97.44
Rectal Temperature (F)				
Level before heat exposure	98.51	98.31	98.20	98.10
Minimum level in 10 min	98.39	98.19	98.03	97.93
Decrease in 10 min	0.13	0.12	0.17	0.17
Level at 1 hr	98.77	98.61	98.61	98.49
Level at 2 hrs	99.40	99.06	99.20	99.00
Thermal Sensations				
Before heat exposure	4.0	4.0	4.1	4.2
On entrance to hot room	6.1	6.3	6.6	6.6
After 1 hr	6.6	6.9	6.3	6.1[a]
After 2 hrs	6.6	6.9	6.2	6.2[a]
Water Losses				
Rate of water loss before heat exposure (Btu per (hr) (sq ft))	5.42	4.93	6.05	5.72
Time of visible sweat (min)	19.6	15.6	14.4	15.8
First hour rate of evaporation in hot room (Btu per (hr) (sq ft))	4.90	6.17	5.46	7.07
Second hour rate of evaporation in hot room (Btu per (hr) (sq ft))	11.85	10.20	13.90	13.94[a]
Sweat in 1 hr (oz)	4.82	3.67	5.43	3.14
Sweat in 2 hrs (oz)	10.41	9.87	12.11	10.22

[a] Effects of clothing are statistically significant.
[b] Effects of relative humidity are statistically significant.

that of the women wearing union suits. This was true in both low and high relative humidity conditions. The women stabilized with significantly lower skin temperatures and lower rates of evaporative heat loss than did the men, Table 3. The thermal sensations of the women were the same whether dressed in union suits or summer clothing and whether exposed to either dry or moist environmental conditions.

Heat Exposure

Immediately on entering the hot room there was the prompt rise in mean skin temperature and the slight initial decline in rectal temperature shown in Fig. 1 and Table 2 similar to the changes observed in men. The increase in mean skin temperature is graphically shown in Fig. 1 as a function of time. The promptness with which temperatures rise is noteworthy. Fig. 1 and Fig. 2 show that, regardless of the costume, the skin temperatures reached higher levels in 10

TABLE 3—COMPARISON OF PHYSIOLOGICAL RESPONSES OF MEN AND WOMEN DRESSED IN UNION SUITS WHEN EXPOSED TO A HOT HUMID ROOM FOR ONE HOUR AND FOLLOWED BY EXPOSURE TO A COMFORTABLE ROOM WITH A 30 PERCENT RH AND 76 F

OBSERVATIONS	PHYSIOLOGICAL ADJUSTMENTS	
	Men[a]	Women
Mean Skin Temperature (F)		
Level before heat exposure	92.31	91.57[b]
Increases in 10 min	4.33	5.27[b]
Level after one hour of heat	96.58	97.22
Decreases in 10 min after heat	6.29	6.47
Level after one hour in comf room	89.33	90.18
Rectal Temperatures (F)		
Level before heat exposure	98.30	98.41
Decreases in 10 min	0.13	0.11
Level after one hour of heat	98.70	98.74
Increase in 10 min after heat	0.14	0.26
Level after one hour in comf room	98.23	98.59
Thermal Sensations		
Before heat exposure	4.1	4.0
On entrance to hot room	6.0	6.1
After one hour of heat	6.3	6.7
On entrance to comfortable room	1.7	2.1
After one hour in comfortable room	4.0	3.9
Water Losses		
Rate before heat exposure (Btu per (hr) (sq ft))	5.79	4.97[b]
Time of visible sweat in hot room min	9.9	19.8[b]
Rate during hour of heat exposure (Btu per (hr) (sq ft))	12.66	5.20[b]
Amount of Sweat (oz)	8.76	4.83[b]
Accumulated sweat from heat exposure (oz per sq ft)	0.30	0.23
Rate of Evaporation in comfortable room after heat exposure (Btu per (hr) (sq ft))	13.72	13.67

[a] Data from reference (1) for men.
[b] Observations for the men and women are statistically significantly different.

min when the women came from a 30 percent than from an 80 percent RH room. Individuals who had higher skin temperatures in the comfortable room also reached higher temperatures in the hot room. The significant factor appears to be the relative humidity of the comfortable room from which the subjects came as indicated in Fig. 2 by the separate lines for each humidity condition of previous exposure.

Immediately after entrance to the hot room there was an increase in weight of the subjects. The technical difficulties involved in accurately following these weight gains prevented a comparison of the gains of subjects coming from the 30 and 80 percent RH environments. It would appear that the increase in weight represented condensation of water in the clothing and that the subject coming from the 30 percent RH environment should show greater gains.

Visible sweat on the face, neck or back of hands of the women was noted on the average in about 20 min. This is markedly later than for the men, see Table 3. Sweating per unit area was also less than for the men. The reader will note

that the levels of skin temperatures attained in 10 min were the same for the women and for the men. The increases in 10 min were greater for the women than for the men and were due to their lower skin temperatures in the comfortable room, see Table 3.

The maximum skin temperatures of the women, regardless of costume, were attained in 1 hr and maintained throughout the second hour. The rectal tem-

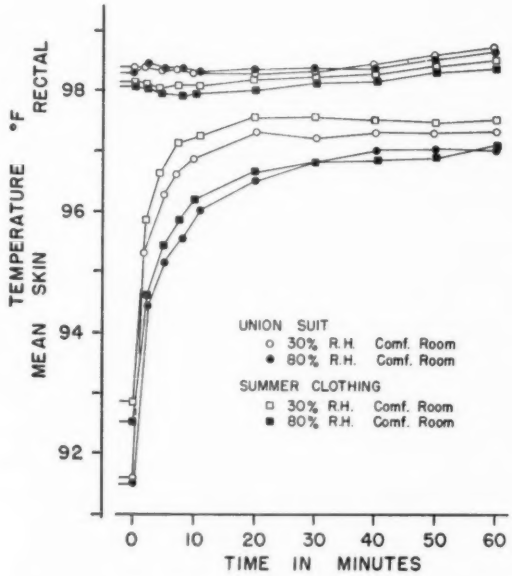


FIG. 1. RESPONSES OF SKIN AND RECTAL TEMPERATURES UPON EXPOSURE TO A HOT HUMID ENVIRONMENT OF WOMEN DRESSED IN SUMMER CLOTHING OR UNION SUIT WHEN PREVIOUSLY EXPOSED TO A 30 AND 80 PERCENT RH OF THE COMFORTABLE ROOM

peratures rose 0.2 to 0.4 F deg in the first hour and an additional 0.5 to 0.6 F deg in the second hour. The continued elevation of rectal temperatures indicated further heat storage. The thermal sensations were the same throughout both hours, and apparently were not influenced by the amount of heat stored. The gradient attained between rectal and mean skin temperatures was 1.5 F deg. These observations are similar to those reported for men subjected to the same experimental conditions, see Table 3.

On Return to the Comfortable Room

On re-entry into the comfortable room there was a sudden sensation of coolness; the skin of the face and neck became dry rapidly. The thermal sensation

returned slowly to a normal range. The observations on the physiological adjustments of the women dressed in either union suits or summer clothing within the comfortable rooms (30 and 80 percent RH) are summarized in Table 4. From a study of Fig. 3 it will be noted that the skin temperatures manifested rapid decreases, which were essentially completed by the end of 10 min, and further slow falls throughout the remainder of the hour. It will be noted,

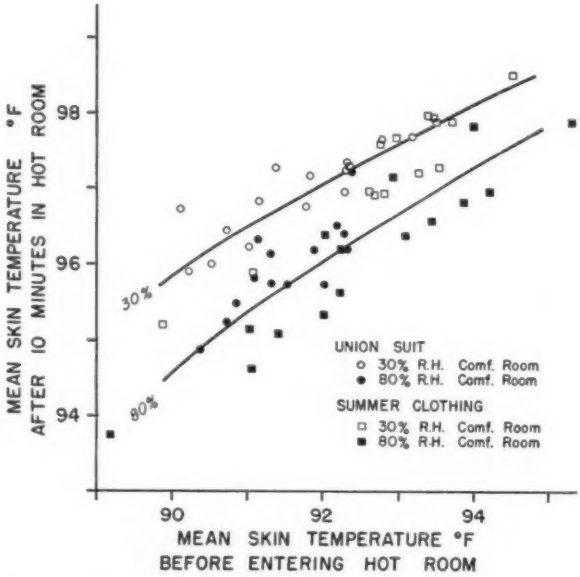


FIG. 2. RELATIONSHIP OF THE LEVEL OF MEAN SKIN TEMPERATURE AFTER 10 MIN IN HOT ROOM TO LEVEL OF MEAN SKIN TEMPERATURE BEFORE ENTERING HOT ROOM WHEN WOMEN WERE DRESSED IN EITHER UNION SUIT OR SUMMER CLOTHING AND PREVIOUSLY EXPOSED TO EITHER A 30 OR 80 PERCENT RH COMFORTABLE ROOM

from Fig. 3 and Table 4, that during the first 10 min greater decreases occurred in the 30 percent RH in contrast to an 80 percent RH environment with women dressed in union suits and in summer clothing. Further, greater decreases occurred in women dressed in union suits as compared to those dressed in summer clothing when exposed to similar relative humidity environments.

During the first 10 min the rectal temperatures and the thermal sensations experienced by the women were the same, regardless of the relative humidity or the costume.

In the remaining 50 min the skin temperature drops were slower, and appeared to be related to the quantity of accumulated sweat, the evaporation of which

tended to delay the rise in skin temperatures. The rectal temperatures were similar with the two types of clothing and with either low or high relative humidity environments (see Table 4). These points will be discussed more completely in a subsequent paragraph. Nevertheless, the women felt nearly comfortable by the end of the hour in each experimental condition.

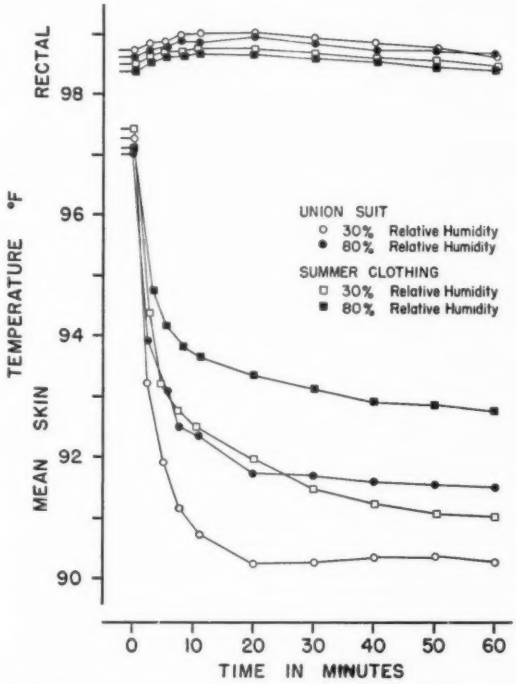


Fig. 3. Responses of Skin and Rectal Temperatures After 1 hr Heat Exposure of Women Dressed in Either Union Suit or Summer Clothing and Exposed to a 30 or 80 Percent RH Comfortable Room

When comparing the adjustments of men and women returning to a comfortable room with clothing saturated with sweat, it is noted in Table 3 that the cooling of the skin, the thermal sensation, and the evaporation per unit area were the same for women and men.

Interpretation of Data (With Respect to Relative Humidity)

On entering the hot room the major physical causes for the rapid rise in skin temperatures were the condensation of moisture on the clothing and the addition

TABLE 4—OBSERVATIONS ON PHYSIOLOGICAL ADJUSTMENTS OF WOMEN CLOTHED IN EITHER UNION SUIT OR SUMMER CLOTHING EXPOSED TO A COMFORTABLE ENVIRONMENT OF EITHER 30 OR 80 PERCENT RH AFTER HAVING BEEN EXPOSED TO A HOT HUMID ENVIRONMENT

OBSERVATIONS	PHYSIOLOGICAL ADJUSTMENTS TO RAPID COOLING							
Type of Clothing	Union Suit				Summer Clothing			
Relative Humidity of comfortable room	30 percent		80 percent		30 percent		80 percent	
Duration of heat exposure	1 hr	2 hrs	1 hr	2 hrs	1 hr	2 hrs	1 hr	2 hrs
Mean Skin Temperature (F)								
Level at end of hot room	97.22	97.82	96.97	97.52	97.14	97.57	97.09	97.44
Decrease in 10 min	6.47	6.35	4.64[b]	4.66[b]	4.63[a]	5.01[a]	3.36[ab]	3.55[ab]
Level at 1 hr	90.18	89.59	91.45[b]	90.87[b]	91.01	90.05	92.78[ab]	92.09[ab]
Rectal Temperature (F)								
Level at end of hot room	98.74	99.40	98.64	99.06	98.53	99.20	98.39	99.00
Maximum increase in 10 min	0.26	0.13	0.23	0.19	0.28	0.13	0.29	0.13
Level at 1 hr	98.59	98.91	98.60	98.64	98.45	98.74	98.40	98.57
Thermal Sensations								
On entrance to comfortable room	2.1	2.1	2.1	2.6	1.6	1.8	2.0	2.0
After 1 hr	3.9	3.8	3.9	3.9	3.9	3.8	4.0	4.0
Water Losses								
Accumulated sweat (oz)	4.09	5.73	4.20	6.25	4.55	8.20	4.03	7.00
Evaporation in 1 hr (Btu per (hr) (sq ft))	12.42	14.46	8.63[b]	10.93[b]	12.38	14.55	9.94[b]	10.47[b]

[a] The effects of different types of clothing are statistically significant.
[b] The effects of relative humidity are statistically significant.

TABLE 5—OBSERVATIONS ON PHYSIOLOGICAL ADJUSTMENTS OF WOMEN DRESSED IN EITHER UNION SUIT OR SUMMER CLOTHING TO SUDDEN CHANGES FROM A HOT HUMID ENVIRONMENT TO A COMFORTABLE ENVIRONMENT ARE COMPARED BY GROUPING DATA INTO LOW AND HIGH EVAPORATIVE HEAT LOSSES

OBSERVATION	PHYSIOLOGICAL ADJUSTMENTS TO COOLING AFTER HEAT EXPOSURE			
Type of Clothing	Union Suit		Summer Clothing	
Average Evaporative heat loss (Btu per (hr) (sq ft))	9.3	15.6	9.3	14.5
Mean Skin Temperature (F)				
Level at end of heat exposure	97.17	97.66	97.25	97.35
Decrease in 10 min	4.94	6.01[b]	3.53[a]	4.47[ab]
Level in 1 hr	91.27	89.88[b]	92.54[a]	91.09[ab]
Rectal Temperatures (F)				
Level at end of heat exposure	98.71	99.26[b]	98.57	98.86[a]
Maximum increase in 10 min	0.22	0.18	0.25	0.20
Level after 1 hr	98.49	98.87[b]	98.42	98.61
Thermal Sensations				
On entrance to comfortable room	2.4	2.1	2.1	1.7
After 1 hr	3.9	3.8	4.0	3.6
Water Losses				
Accumulated sweat (oz)	3.69	6.70[b]	4.74	7.22[b]

[a] Effects of clothing are statistically significant.
[b] Effects of evaporative heat losses are statistically significant.

of heat by radiation and convection. The thermal effects of condensation are opposite to those due to the evaporation of sweat. Dry clothing exposed to an environment of low relative humidity, in contrast to the same clothing exposed to an environment of high relative humidity, permits greater condensation of atmospheric moisture on entering a hot humid environment.

The effect of relative humidity on the physiological adjustments to sudden cooling was clearly shown in Fig. 3. A 30 percent, in contrast to an 80 percent RH environment, caused the faster and greater decrease of mean skin tempera-

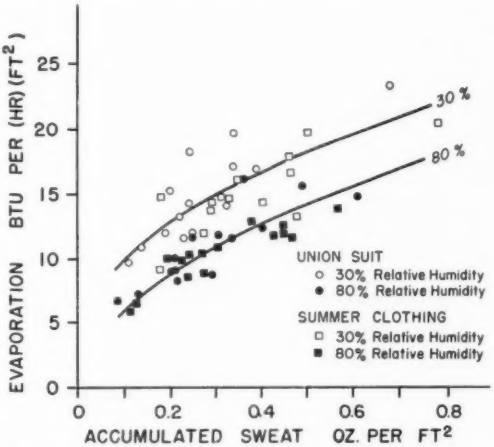


FIG. 4. RELATIONSHIP OF ACCUMULATED SWEAT TO ITS EVAPORATION FOR WOMEN DRESSED IN UNION SUIT OR SUMMER CLOTHING AND EXPOSED TO A 30 AND 80 PERCENT RH COMFORTABLE ROOM

tures of similarly clothed women, Table 4. This 50 percent difference in relative humidity was responsible for the divergence in skin cooling. The manner in which relative humidity exerted its effect was through its ability to regulate the rate of evaporation of the pool of accumulated sweat, see Fig. 4. It was this evaporative cooling which immediately determined the course of the rapid adjustments. Thus, as early as 10 min, a 15.6 Btu per (hr) (sq ft) evaporative heat loss caused a greater decrease in mean skin temperature of similarly clothed women than a 9.3 Btu per (hr) (sq ft) loss, Table 5 and Fig. 5.

Evaporation supplements cooling by radiation and convection. In the absence of evaporation, however, heat loss by radiation and convection would be the same in either low or high relative humidity environments. Relative humidity indirectly affected heat loss by radiation and convection of similarly clothed subjects by altering the temperature gradient between clothing and walls, see Fig. 3. Increases in rates of heat loss by evaporation, upon exposure to a low as compared to a high relative humidity environment, cause reduction in heat loss by radiation and convection.

The relative humidity, in the presence of an adequate supply of accumulated sweat, determines the course of skin temperatures after the completion of the initial rapid adjustments. During this period of 50 min, its influence was exerted on evaporation; the greater evaporative heat loss caused a greater skin temperature drop during the hour as shown in Fig. 6. For complete evaporation of the sweat, greater quantities of heat from the body were required and necessi-

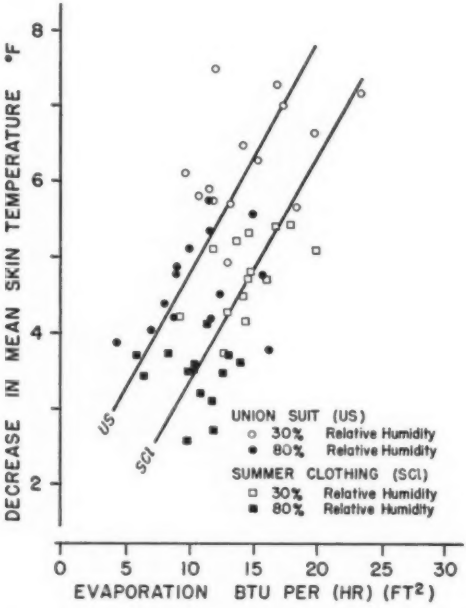


FIG. 5. EFFECTS OF EVAPORATION ON DECREASE IN MEAN SKIN TEMPERATURE IN 10 MIN OF WOMEN DRESSED IN EITHER UNION SUIT OR SUMMER CLOTHING AND EXPOSED TO EITHER 30 OR 80 PERCENT RH COMFORTABLE ROOM

tated prolonged periods of lowered skin temperatures. Excessive body cooling by rapid evaporation was prevented by the vaso-motor mechanisms. This is shown by the maintenance, during the entire hour, of the same rectal temperatures.

INTERPRETATION OF DATA (WITH RESPECT TO CLOTHING)

The mean skin temperature in the comfortable room was higher (statistically significant) with summer clothing than with union suits. This is explained by the greater insulation offered by the summer clothing. The lesser increments in skin temperature on entering the hot room are, therefore, to be expected since

all subjects stabilized at the same skin temperature in the hot room as shown in Table 2.

The effect of the two types of clothing was clearly shown in Fig. 3 when the women were exposed to the comfortable room after heat exposure. The summer clothing, in contrast to the union suit, decreased the rate of skin cooling

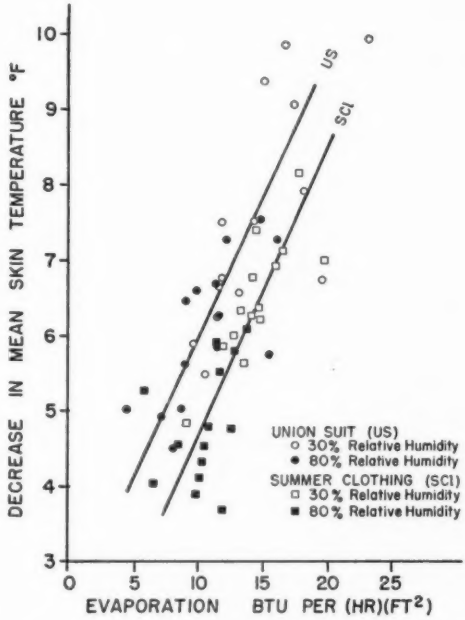


FIG. 6. EFFECTS OF EVAPORATION ON THE DECREASE IN SKIN TEMPERATURE AFTER 1 HR IN A COMFORTABLE ROOM WITH EITHER A 30 OR 80 PERCENT RH. THE WOMEN WERE DRESSED IN EITHER UNION SUIT OR SUMMER CLOTHING

of women exposed to similar environments, Table 5. The differences in the types of clothing are responsible for the diverging skin temperature curves.

At this point it should be well to indicate the physical differences in the two types of clothing. The summer clothing offered a triple layer insulation with air spaces between layers; the union suit offered a single layer insulation. Both permitted free passage of water vapor.

The effects of clothing were further demonstrated by relating evaporative heat loss for the hour in each subject to the decrease in mean skin temperature in 10 and 60 min as shown in Figs. 5 and 6, respectively. Each type of clothing had

its own regression line; these were found to be significantly separate upon statistical analysis. It can then be said that the summer clothing, in contrast to the union suit, permitted less decrease in skin temperature when the evaporative heat losses were the same. This is also shown in Table 5. The effects were observed as early as 10 min and maintained throughout the hour.

The summer clothing required the body to maintain higher skin temperatures to force through the clothing the same amount of radiative and convective heat which is lost through the union suit under circumstances of equal rates of evaporative heat loss. The differences in skin temperature after complete evaporation of the accumulated sweat should be maintained or should approach those differences existing before heat exposure.

INTERPRETATION OF DATA (WITH RESPECT TO SEX)

The literature contains few references to studies of heat loss in women[3] although their lower heat production (basal metabolism) has been extensively studied. In terms of adaptability to sudden heat exposure, women can reach the same maximal skin temperatures just as promptly as men, even though they start from lower levels. This response is subject to further study. The slowness of initiating sweat in women had been previously reported[4]. In this regard the data compiled by the authors confirm those of DuBois[4]. The lower rate of sweating in women as compared to men is also confirmed.

The thermal sensations showed that the sexes adjust with equal promptness. Within the experimental ranges studied, the behaviors of the sexes were quite similar and indicate that the presence of men and women in the same space does not require alterations in accepted specifications for its air conditioning. The similarities in the other observations of men and women subjected to rapid cooling are perhaps explained by the high rates of evaporative cooling. This is also under further study in view of the well established fact that the higher fat content in the body of women[5] may be physiologically active as an insulating mechanism during exposures to cold environments.

CONCLUSION

Women adjust as readily as do men to sudden entrance into a hot humid environment. Their skin temperatures rise promptly, even faster than is observed for men. Sweat is slower to appear than in the case of the men.

As women leave a hot humid room and enter one which is normally comfortable, they feel cool, stop sweating promptly, and rapidly adjust their skin temperatures much as do men. This adjustment is controlled by: (1) sweat accumulated in their clothing and available for evaporative heat loss; (2) relative humidity; and (3) clothing.

REFERENCES

1. Physiological Adjustments of Clothed Human Beings to Sudden Change in Environment—First Hot Moist and Later Comfortable Conditions, by F. K. Hick, M.D., Tohru Inouye, R. W. Keeton, M.D., Nathaniel Glickman and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, pp. 189-198).

2. A.S.H.V.E. RESEARCH REPORT No. 1320—Physiological Adjustments of Human Beings to Sudden Changes in Environment, by Nathaniel Glickman, Tohru Inouye, S. E. Telser, M.D., R. W. Keeton, M.D., and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, pp. 327-356).

3. Basal Heat Loss and Production in Women at Temperatures from 23 C. to 36 C., by J. D. Hardy and A. T. Milhorat (*Proceedings of The Society for Experimental Biology and Medicine*, Vol. 41, 1939, pp. 94-98).

4. Differences Between Men and Women in their Response to Heat and Cold, by J. D. Hardy and E. F. DuBois, M.D. (*Proceedings of the National Academy of Sciences*, Vol. 26, 1940, pp. 389-398).

5. A Method of Breaking Down the Body Weights of Living Persons into Terms of Extracellular Fluid, Cell Mass and Fat, and Some Applications of It to Physiology and Medicine, by R. A. McCance and E. M. Widdowson (*Proceedings of the Royal Society*, Series B, Number 890, Vol. 138, 1951, pp. 115-129).

DISCUSSION

LESTER T. AVERY, Cleveland, Ohio: This is one of a series of physiological research projects which this Society has sponsored for a number of years. I think it is of tremendous importance. Ten or fifteen years ago, we were very much concerned with what was alleged to be the ill effects of shock, the strain on the individual. These experiments and others like them show that this idea had no physiological basis and that the human body could make adjustments when necessary without strain.

I raise one question here, because it is so foreign to our own experience, and that is whether one could have a tolerable feeling of comfort with 80 percent RH (relative humidity) at a temperature of 76 F. If these men report such a finding it must be so. It is not common to our experience because people are not bound in swaddling clothes and kept completely at rest.

The people with whom we work in normal life are carrying on some alleged activity, some being more vigorous than others. Therefore we cannot find any place in our own experience where 80 percent RH and 76 F dry-bulb temperature, which will be somewhat above 73 ET (effective temperature) would be tolerable for anybody.

AUTHORS' CLOSURE (Professor Glickman): This paper presented by Mr. Inouye, shows a few of the physiological differences between men and women in their *response* to temperature and humidity changes. It confirms some of the findings reported by Hardy and DuBois in 1940; by Yaglou and Messer in 1941 and amplifies a statement made by Houghten, Giesecke and Tasker in 1937 in a paper presented before this Society. In one part of the paper they mentioned *comparing* the subjective responses of men and women. They reported that *Few data were obtained for women of different age groups under satisfactory weather and environmental conditions, but these few observations indicate no material difference in the degree of cooling desired over that found for men.*

The reason the women have a lower mean skin temperature than the men at the *lower ambient* temperatures, as shown here, is that they have a greater amount of subcutaneous fat which *affords* them a greater *tissue insulation.* However, the interesting thing, to *me*, as shown by Hardy and DuBois, is that at the *higher* ambient temperatures the women were able to *reduce* their tissue insulation and have a higher mean skin temperature than the men, as was also shown here.

This paper just about completes one phase of a series of experiments. The purpose of the first phase was to determine the physiological and subjective responses of healthy individuals to sudden changes in ambient temperature and humidity. The second phase was to extend this to impaired patients.

Just what have we learned from this phase of the experiments and how can we apply this knowledge? Briefly, we have learned that both *men* and *women* are able to adjust rapidly and with little strain on the cardiovascular system, *i.e.*, heart and circulation. Their adjustments are similar in most respects, except for mean skin temperatures and evaporative weight loss.

Clothing alters the magnitude of the responses and it alters the requirements for comfort particularly at the lower temperatures (in the zone of body cooling).

We found *no* evidence of adaptation to cold when we compared the responses of the *same* subjects in *summer* and *winter*.

The observed responses of older patients with heart disease were similar in most respects to those of young normal subjects.

We can apply these findings by citing that entering an air conditioned space from the hot outside and, conversely, entering the hot outside from an air conditioned space, is not harmful. However, by this I do not mean to imply that all diseased states would respond with so little evidence of stress.

The term *Shock* as applied to entering and leaving air conditioned space really should no longer be used by engineers. It connotes something far different in medical terminology.

There is one observation that disturbs me and it is a point that indicates the need for further study. In the comfortable room, before entering the hot room, the subjects indicated they were comfortable regardless of whether they were dressed in the union suit or in the summer clothing which for the men included a regular suit, tie and shirt buttoned at the neck. It seems to me that for the men in summer clothing the ambient conditions used were on the borderline of the zone of evaporative regulation. If they had engaged in very light standardized activity instead of sitting quietly as they did, I believe they would have been slightly warm, *i.e.*, they would have been moved into the zone of evaporative regulation whereas the subjects in the union suits may have remained comfortable. This indicates the need for further studies with clothed subjects engaged in light activity (I may add that I am aware of the earlier studies conducted at the Society's Laboratory when it was in Pittsburgh).

1464

No. 1464

TWO METHODS OF CALCULATING THE FRICTION IN PIPE LINES

By F. E. GIESECKE*, NEW BRAUNFELS, TEX.

A PIPE line consists of sections of straight pipe, elbows, and other single resistances; hence, to calculate the friction in a pipe line, it is necessary to know how to calculate the friction in a straight pipe and also how to calculate the friction in elbows and in other single resistances.

The friction in new and clean, black steel or wrought-iron, standard weight, pipe can be calculated accurately by means of the chart in THE GUIDE 1952 (p. 513) and also by means of similar charts and tables in other engineering books. Accurate charts and tables for use in calculating the friction heads in elbows and other single resistances are not now generally available.

In 1917 The University of Texas issued Bulletin No. 1759—The Friction of Water in Pipes and Fittings. This bulletin describes a study conducted by the writer to determine the friction of water in pipes and fittings. The study of the friction in elbows led to the following general formula:

$$h = 0.0141\, d^{-0.26}\, v^{1.96} \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

where

h = friction head in one black iron, standard weight, 90 deg short radius, screwed elbow, feet of water.
d = internal diameter of the pipe, inches.
v = velocity of the water in the pipe, feet per second.

This study also led to the following statement:

For the range of sizes (½ in. to 2½ in.) and velocities (0.56 fps to 2.66 fps) investigated, the friction of a standard 90-deg elbow is about the same as that of a pipe whose diameter is equal to that of the elbow and whose length is equal to 25 diameters.

---

* Consulting Engineer. Life Member and Presidential Member of A.S.H.V.E.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

This statement seems to be the origin of the belief now quite commonly held that the friction in one standard elbow is equal to the friction in a pipe whose nominal diameter is the same as that of the elbow, and whose length is 25 times its internal diameter.

Since the studies at The University of Texas were confined to standard elbows ranging in size from ½ in. to 2½ in. and to water velocities ranging from 0.56 fps to 2.66 fps, the suggestion that a pipe length of 25 diameters be considered equivalent to one elbow should be applied only within those limits. It seems, however, that the suggestion is frequently applied when the elbow is larger

TABLE 1—A COMPARISON OF FRICTION HEADS IN STANDARD ELBOWS AND PIPES

(For 1 and 10 fps Velocity)

Nominal Size	One Foot per Second Velocity				Ten Feet per Second Velocity			
	Friction Head (Milinches)		Equivalent of One Elbow		Friction Head (Milinches)		Equivalent of One Elbow	
	One Elbow	One Foot of Pipe	Feet of Pipe	No. of Pipe Diameters	One Elbow	One Foot of Pipe	Feet of Pipe	No. of Pipe Diameters
½	182	150	1.21	23	18,200	10,900	1.67	32
1	170	76	2.24	26	17,000	5,200	3.27	37
2	147	32	4.59	27	14,700	2,100	7.00	41
3	132	20	6.60	26	13,200	1,240	10.65	42
4	120	14	8.57	26	12,000	870	13.79	41
6	103	8.4	12.26	24	10,300	500	20.60	41
8	90	6.0	15.00	23	9,000	350	25.71	39
10	82	4.4	18.64	22	8,200	250	32.80	39
12	78	3.8	20.53	21	7,800	210	37.14	37
			Average 24.22				Average 38.77	

than 2½ in. and when the velocity is higher than 2.66 fps. For example, in THE GUIDE, 1952 (p. 485), it is stated that the friction in 17 ft of 8-in. pipe is equivalent to the friction in a standard 8-in. elbow, whereas, according to Table 1, the *equivalent length* of an 8-in. elbow ranges from 15.00 ft to 25.71 ft as the velocity ranges from 1 fps to 10 fps.

In 1892 J. R. Freeman, civil engineer, conducted a very extensive series of experiments in his New Hampshire laboratory to determine the friction of water flowing in pipes and fittings. The results of this classic study had been dormant for nearly 40 years when they were published by Mr. Freeman's son, Clarke, assisted by Dr. C. P. Kittredge and others.†

To determine the friction in standard elbows, Mr. Freeman performed 81 experiments. He experimented with elbows ranging in size from 2½ in. to 8 in.

† *Flow of Water in Pipe and Pipe Fittings*, by John Ripley Freeman, C.E. (published by *The American Society of Mechanical Engineers*, 1941, 350 pp.).

and with velocities ranging from 0.6 fps to 23.5 fps and found the following average friction heads per elbow:

2½-in. elbow, 12 experiments, $h = 0.772\ v^2/2g$ ft or $144\ v^2$ milinches.
3-in. elbow, 12 experiments, $h = 0.742\ v^2/2g$ ft or $138\ v^2$ milinches.
4-in. elbow, 28 experiments, $h = 0.631\ v^2/2g$ ft or $118\ v^2$ milinches.
6-in. elbow, 22 experiments, $h = 0.574\ v^2/2g$ ft or $107\ v^2$ milinches.
8-in. elbow, 7 experiments, $h = 0.457\ v^2/2g$ ft or $85\ v^2$ milinches.

where

h = friction head of water per elbow, feet or milinches.
v = mean velocity of the water, in the pipe, feet per second.

Mr. Freeman's values of the friction heads in standard elbows, ranging in size from 2½ in. to 8 in., expressed in the form $h = Kv^2$ milinches and the writer's

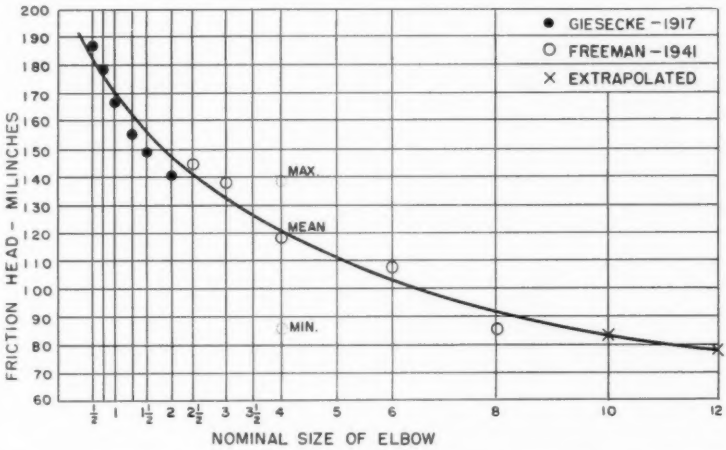


FIG. 1. FRICTION HEAD IN ELBOWS FROM GIESECKE AND FREEMAN TESTS

For water flowing through a standard elbow with a velocity in the pipe of one foot per second.

values of the friction heads in standard elbows, ranging in size from ½ in. to 2½ in., calculated by means of Equation 1, are shown in Fig. 1 for the velocity of 1 fps.

The curved line in Fig. 1 shows the average results of the two investigations; it was used to determine the friction head coefficients for elbows of various sizes as shown in Table 2.

In considering the possible inaccuracy of this graphical method of finding the friction heads in standard elbows, it should be borne in mind that it is much more difficult to determine the friction heads in elbows than it is to determine the friction heads in 1 ft of straight pipe because the friction in an elbow is affected by the distance the pipe is screwed into the elbow, by the turbulence in the water when it enters the elbow, and by the roughness of the internal

surface which may be different in elbows produced by different manufacturers. The difficulty of determining the friction head in an elbow is illustrated by Mr. Freeman's experimental study of the friction in a 4-in. elbow. His study included three sets of experiments: in the first set were 4 elbows, 20 ft apart, and 9 experiments; the average value of K (in $h = Kv^2/2g$) was 0.6251. In the second set were 4 elbows, 0.3 ft apart, and 12 experiments; the average value of K was 0.6981. In the third set were 4 elbows, 0.3 ft apart, and 7 experiments: the average value of K was 0.5348. The average value of K for the 28 experiments was $0.631v^2/2g$ ft or $118v^2$ milinches as shown in Fig. 1. The minimum of the 28 experimental values was 84 and the maximum value was 138 as shown by the two dotted circles in Fig. 1.

The corresponding friction heads, per foot of pipe, shown in Table 1, were taken from the chart on page 513 of THE GUIDE 1952. The friction heads shown in that chart were calculated by means of the equation

$$h = 81d^{-1.26}\, v^{1.84d^{-0.018}} \text{ milinches}$$

Knowing the friction heads per elbow and per foot of pipe, the *equivalent lengths* of pipe per elbow were calculated and expressed in feet of pipe and in pipe diameters, as shown in Table 1.

TABLE 2—FRICTION HEAD COEFFICIENTS FOR STANDARD ELBOWS[a, b]

NOMINAL SIZE INCHES	VALUE OF K	NOMINAL SIZE INCHES	VALUE OF K	NOMINAL SIZE INCHES	VALUE OF K
½	182	2	147	5	111
¾	176	2½	140	6	103
1	170	3	132	8	90
1¼	162	3¼	127	10	82
1½	156	4	120	12	78

[a] A standard elbow is a black steel or wrought-iron, 90 deg, short radius, screwed elbow.
[b] For use in formula: $h = Kv^2$; h = milinches; v = feet per second.

It appears from Table 1 that, for a velocity of 1 fps, the *equivalent length* of pipe per elbow, ranging in size from ½ in. to 12 in., expressed in pipe diameters, ranges from 21 to 27 with the average of 24.2 and that, for the velocity of 10 fps, the range is from 32 to 42 with the average of 38.8.

Since velocities used in industry are generally higher than 1 fps, friction head calculations for pipe lines, based on the assumption that the *equivalent length* of the pipe is 25 times its internal diameter, are only approximately correct; hence this method of calculating friction heads may be called the *approximate method*. The name *correct method* may be used when separate friction head calculations are made for the pipe and for the elbows and other single resistances, and the two calculated friction heads added to secure the final result.

Among the single resistances used in pipe lines, other than the standard elbow, are couplings, tees, valves, and radiators. The friction in these units

is generally expressed in terms of the friction in a standard elbow, *i.e.,* in elbow equivalents. For example, the elbow equivalent of an open gate valve is considered to be 0.5; this value was determined for low velocites and small sizes. Since the friction in the single resistances varies practically as the square of the velocity, the elbow equivalent of a single resistance is the same for all velocities. This differs from the relation between the friction in a pipe and that in an elbow because the friction in a pipe varies practically as the 1.80 power of the velocity whereas the friction in an elbow varies practically as the square of the velocity; hence, if the *equivalent length* of an elbow is 25 pipe diameters for some one velocity it is more than 25 diameters for every higher velocity and less than 25 diameters for every lower velocity.

Whether the elbow equivalent of an open gate valve is 0.5 for all sizes of gate valves must be determined experimentally as was done for elbows by Mr. Freeman. Until this has been done it is necessary to assume that the number of elbow equivalents is not affected by the size of the fitting. In the case of gate valves it is safe to do so because there is generally not more than one gate valve in a pipe line.

The difference between the approximate method and the *correct* method of calculating friction can be found by calculating the friction heads in the pipe

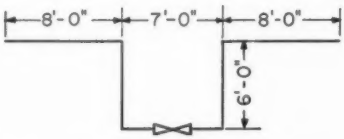


FIG. 2. PIPE LINE FOR ILLUSTRATION OF FRICTION LOSS CALCULATIONS

line shown in Fig. 2 for velocities of 1 fps and 10 fps. The pipe line consists of 35 ft of 4-in. pipe, 4 elbows, and one gate valve, or of 35 ft of pipe and 4.5 elbow equivalents.

For a velocity of 1 fps:

By the approximate method the equivalent length of the pipe line is $35 + 4.5 \times 25 \times 4.026/12$ or 72.7 ft and the friction head is $72.7 \times 14 = 1018$ milinches.

By the correct method the friction head in the pipe is 35×14 or 490 milinches and in the five single resistances (4.5 elbow equivalents) it is 4.5×120 or 540 milinches, and, together, $490 + 540 = 1030$ milinches, a difference of only one percent.

This shows that, for a 4-in. pipe and a velocity of 1 fps, the approximate method gives results which are practically correct.

For a velocity of 10 fps:

By the approximate method, the equivalent length of the pipe line is as before 72.7 ft; the friction head is $72.7 \times 870 = 63{,}249$ milinches or 5.27 ft.

By the correct method, the friction head in the pipe is $35 \times 870 = 30{,}450$ milinches and in the five single resistances it is $4.5 \times 120 \times 100 = 54{,}000$ milinches and the total is $30{,}450 + 54{,}000 = 84{,}450$ milinches or 7.04 ft.

According to these calculations the friction head calculated by the approximate method is 25 percent lower than that calculated by the correct method. So great an inaccuracy is not permissible in good engineering design and, since the correct method is practically as simple as the approximate method, it is best to abandon the use of the approximate method entirely.

1465

No. 1465

COMPRESSION TANK SELECTION FOR HOT WATER HEATING SYSTEMS

By H. A. LOCKHART* AND G. F. CARLSON**, MORTON GROVE, ILL.

CURRENTLY ACCEPTED methods for sizing compression tanks give widely divergent results. For instance, on a specific single story heating system, the size of compression tank installed could be 11 gal, by a formula in HEATING VENTILATING AIR CONDITIONING GUIDE 1952, Chapter 21—Hot Water Heating Systems, or 35 gal by *A.S.M.E.* Code requirements. Other sizing methods would yield still different results. The size of compression tank installed today depends as much on the tank sizing method used as on the particular system.

Compression tanks are used in hot water heating systems for three reasons: (1) to allow the system to operate at temperatures above 212 F; (2) to control rate of pressure increase with temperature rise in the system; and (3) to eliminate relief valve operation, with consequent need for refilling the system with water, unless unsafe conditions occur.

Chapter 21 of THE GUIDE 1952, establishes a more exact tank size than the empirical methods otherwise employed, as it is based on actual water content. The simplicity of empirical methods, however, often leads designers to use them rather than the formula in THE GUIDE 1952 which requires more work in determination of system water content.

It is the purpose of this paper to present a safe and rational compression tank sizing method which will be basically as simple as the present empirical methods. Table 1 shows a tank selection chart prepared for this purpose.

The upper section of Table 1 shows the maximum tank size required separately for the boilers, radiation, forced circulation piping, and for piping and panels in a panel heating system. Maximum tank size for a complete system is found by adding the maximum tank capacities required for each component of the system. Maximum tank capacity requirements are based on 22 percent of the average water contents of boilers, radiation and piping, and adopted standard

* Chief Engineer, Bell & Gossett Co. Member of A.S.H.V.E.

** Engineering Department, Bell & Gossett Co.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

TABLE 1—PROPOSED METHOD FOR SELECTING PROPER SIZE OF COMPRESSION TANK

Maximum Tank Size Required For Boiler (Under Standard Conditions)[a]					Maximum Tank Size Required For Radiation (Under Standard Conditions)[a]									Maximum Tank Size Required for Forced Circulation Piping (Under Standard Conditions)[a]					Maximum Tank Size Req'd For Panel (Under St'd Conditions)[a]			
Net Btu Rating in 1000 Btuh	Conventional		Flash		Rad Output in 1000 Btuh at 200 Btuh Emission	Large Tube		Thin Tube		Convector		Base Board		Heat Conveyed in 1000 Btuh	One Pipe		Two Pipe		System Btu in 1000 Btuh	Panel	
	Gals in Boiler	Tank Req'd	Gals in Boiler	Tank Req'd		Gals in Rad	Tank Req'd	Gals in Rad	Tank Req'd	Gals in Rad	Tank Req'd	Gals in Rad	Tank Req'd		Gals in Piping	Tank Req'd	Gals in Piping	Tank Req'd		Gals in Panel Piping	Tank Req'd
50	12	2.64	5.4	1.19	50	28.5	6.27	15.7	3.4	7	1.5	3.2	0.704	50	7.8	1.7	10.5	2.3	50	11	0.8
60	14.5	3.19	6.4	1.41	60	34.2	7.52	16.8	3.7	7.5	1.65	4.0	0.880	60	9.0	1.98	12.5	2.75	60	13.5	1.0
70	17	3.74	7.2	1.58	70	40	8.8	19.6	4.3	8.7	1.93	4.5	0.990	70	10.0	2.2	14.5	3.2	70	16	1.2
80	19.5	4.29	8.0	1.76	80	45.6	10	22.4	4.9	10	2.2	5.2	1.14	80	11.5	2.5	17	3.7	80	18.5	1.35
90	22	4.9	8.6	1.89	90	51.4	11.3	25.2	5.5	11.25	2.48	5.8	1.27	90	13	2.9	18	3.96	90	21	1.55
100	25	5.5	9.4	2.07	100	57	12.5	28	6.2	12.5	2.75	6.5	1.43	100	14	3.1	21	4.6	100	24	1.75
125	30	6.6	11	2.42	125	71	15.6	39	8.58	17.7	3.89	8.1	1.78	125	18	3.96	27	5.9	125	30	2.2
150	36	7.9	13	2.86	150	85.5	18.8	47.8	10.5	21	4.62	9.7	2.13	150	22	4.85	34	7.5	150	37	2.7
175	42	9.24	14.5	3.19	175	99	21.7	55	12.1	24.7	5.43	11.3	2.48	175	26	5.7	40	8.8	175	44	3.2
200	48	10.6	16	3.52	200	114	25.8	63.8	14.1	28.5	6.27	13	2.86	200	30	6.6	47	10.3	200	50	3.7
250	60	13.1			250	142	31.2	79.8	17.5	35.6	7.83	16.2	3.56	250	39	8.6	62	13.6	250	65	4.75
300	70	15.4			300	171	37.6	95.7	21.0	42.7	9.39	19.5	4.29	300	49	10.8	78	17.2	300	70	5.7
350	84	18.4			350	199	43.7	111	24.4	49.7	10.9	22.7	4.99	350	58	12.8	94	20.7			
400	95	20.9			400	228	50.1	127	27.9	57	12.5	26	5.72	400	70	15.4	110	24.2			
450	107	23.5			450	256	56.1	143	31.4	64	14	29.2	6.42	450	80	17.6	130	28.6			
500	120	26.4			500	285	62.7	157	34.5	71	15.6	32.5	7.15	500	90	19.8	150	33			
600	140	30.8			600	342	75.2	191	42	85.5	18.8	39	8.58	600	115	25.3	190	41.8			
700	160	35.2			700	399	87.7	223	49	98.7	21.7	45.5	10	700	140	30.8	225	49.5			
800	190	41.8			800	456	100	251	55.2	114	25	52	11.4	800	165	36.3	260	57.2			
900	210	46			900	513	112	287	63.1	128	28.1	58.5	12.8	900	195	42.9	310	68.2			
1000	235	51.7			1000	570	125	319	70.1	142	31.2	65	14.3	1000	225	49.5	360	79.2			
2000	480	106			2000	1140	250	638	140	285	62.7	130	28.6	2000	540	119	900	198			
3000	720	158			3000	1710	376	957	210	427	93.9	195	42.5	3000	960	211	1500	330			
4000	960	211			4000	2280	501	1276	280	560	123	260	57.2	4000	1350	297	2100	462			
8000	1200	264			8000	4560	1003	2553	561	1140	250	520	114	8000	3400	748	5000	1100			
12000	3000	660			12000	6840	1504	3830	832	1710	376	780	171	12000	5000	1254	9000	1980			
16000	3700	814			16000	9120	2006	5107	1123	2280	501	1040	228	16000	8300	1826	13000	2860			
					20000	11400	2508	6384	1407	2850	627	1300	286								

Note:

For minimum tank size under standard conditions multiply Value obtained from table by 0.73

TABLE 1—PROPOSED METHOD FOR SELECTING PROPER SIZE OF COMPRESSION TANK *(continued)*

Correction Factors For Initial Pressures Other Than 12 Psig (4 Psig Min Pressure Highest Part of System, 30 Psig Relief Pressure)					Correction Factors For Attic Tanks (4 Psig Initial Pressure on Tank Press., Rise = Final Press.-Static-4 Psig)				Correction Factors For Systems Using Compressor to Charge Tank (Comp Tank 80% Full of Air After Charging with Compressor)						Correction Factors For Safety Loop Systems				Correction Factors For Tanks that are Already Undersized	
Initial Pressure	Static Height	Max Mean Temp at Max Mult Factor	Multiplying Factor Maximum	Multiplying Factor Minimum	Pressure Rise	Max Mean Temp at Max Mult Factor	Multiplying Factor Maximum	Multiplying Factor Minimum	Static Height	Initial Pressure	Relief Pressure	Max Mean Temp at Max Mult Factor	Multiplying Factor Maximum	Multiplying Factor Minimum	Initial Pressure	Max Mean Temp at Max Mult Factor	Multiplying Factor Maximum	Multiplying Factor Minimum	If Tank Reaches Final Press. At Temp	Required Tank Size Increase For Final Press. at 220F
4	0	264	0.465	0.3	4	222	1.27	1.25	10	8.5	30	256	0.53	0.35	4	262	0.72	0.36		Orig-
6	4.5	260	0.575	0.38	8	235	0.77	0.68	18.5	12	30	252	0.65	0.48	6	262	0.98	0.56		inal
8	9	257	0.7	0.48	10	238	0.68	0.57	25	15	30	248	0.85	0.67	8	262	1.24	0.74		Tank
10	14	255	0.83	0.59	12	242	0.63	0.50	30	17	30	244	0.97	0.79	10	262	2.16	1.0	170	2.1 × Vol.
12	18.5	252	1.0	0.73	14	246	0.59	0.45	40	21.5	45	255	0.71	0.47	12	262	3.5	1.5	180	1.87 × Vol.
14	23	250	1.19	0.9	16	249	0.55	0.41	50	26	45	251	0.815	0.59					190	1.63 × Vol.
16	27.5	245	1.43	1.1	18	252	0.53	0.38	60	30	45	248	1.04	0.82					200	1.34 × Vol.
18	32	242	1.74	1.41	20	255	0.51	0.36	75	37	75	264	0.68	0.78					210	1.24 × Vol.
20	36.5	237	2.18	1.82	25	261	0.48	0.315	100	47	75	264	0.97	0.62					220	1.13 × Vol.
22	41.5	235	2.82	2.5	30	267	0.465	0.28	150	65	100									
24	46	227	3.82	3.5	35	271	0.45	0.26												

[a] Standard conditions are: 12 psig initial pressure, 30 psig final pressure, static pressure up to 18.5 ft.

conditions of 12 psig initial pressure, 30 psig final pressure and 18.5 ft maximum system height above the tank.

Table 1 lists correction factors for multiplying maximum tank capacities to obtain either maximum or minimum tank capacities for conditions other than standard.

Volume data for boilers, radiation and piping, and operating conditions adopted as standard, and a basis for correction factors for conditions other than standard, are presented in Appendix A.

The majority of hot water heating systems are installed with the tank in the basement, with an initial fill pressure of 12 psig, and with a final relief valve pressure of 30 psig. In this paper, the maximum temperature at which relief pressure is reached is assumed to be 262 F, the minimum relieving temperature being 220 F. A 4 psig fill pressure is established as minimum at the top of the system and the maximum system height for standard conditions is 18.5 ft.

A rational estimate of compression tank volume requirements indicates that the tank should be sized between 16 and 22 percent of the total system volume under standard conditions. Table 1 is based on 22 percent of the volume of any given component of the heating system. *Maximum tank size* for any system is determined by adding the tank requirement of each component; piping, radiation, and boiler. *Minimum tank size* is determined by multiplying maximum tank size by 0.73.

Table 1 establishes minimum tank size at a condition such that relief pressure is reached at 220 F. For systems utilizing a design temperature higher than 220 F the minimum tank size established by the table cannot be used. Use either the maximum tabulated tank size or establish minimum tank size by an equation given in this paper in the section, Formulas for Determining Tank Size.

The *effective size* of a given compression tank is determined by the relation of the volume of air within the tank, under initial fill pressure or cold non-operating conditions, to the volume of water expansion it is expected to absorb. A lowering of the initial fill pressure increases the air volume and hence increases effective tank size. An increase of initial fill pressure, as caused by a high static head system, decreases effective tank size. In order to reduce tank size, it may be necessary to install the tank at the highest point of a high static head system or lower the initial fill pressure below standard on a low static head system.

The following *Examples 1 to 6* will illustrate the method of selecting tank sizes by use of Table 1:

Example 1: Select the size of compression tank for standard conditions for a hot water heating system having: 100,000 Btuh conventional boiler; 90,000 Btuh thin-tube radiation @ 200 Btuh emission = 450 sq ft (water radiation); 90,000 Btuh one-pipe system; 5 ft static head; 12 psig initial fill pressure; 30 psig final pressure; and tank located at boiler.

Solution:

(*a*) Establish maximum tank size from Table 1:

100,000 Btuh conventional boiler requires	5.5 gal
90,000 Btuh thin tube requires	5.5
90,000 Btuh one-pipe system requires	2.9
Maximum tank size	13.9 gal

(*b*) Establish minimum tank size:

$$13.9 \text{ gal} \times 0.73 = 10.1 \text{ gal}$$

(*c*) Tank selection will be made between 10.1 and 13.9 gal. A 12-gal tank is therefore selected.

Example 2: For the conditions of *Example 1,* determine the smallest tank that can be used by lowering the initial pressure.

Solution:

(*a*) Establish lowest possible initial pressure. From Table 1 for initial pressure other than 12 psig, 8 psig can be used for a static head of 5 ft.

(*b*) Maximum and minimum correction factors at 8 psig initial pressure are: for maximum tank size correction factor = 0.7; for minimum tank size correction factor = 0.48.

(*c*) Maximum and minimum tank sizes will then be:

$$\text{maximum} = 13.9 \times 0.7 = 9.75 \text{ gal};$$
$$\text{minimum} = 13.9 \times 0.48 = 6.7 \text{ gal}.$$

(*d*) An 8 gal tank will be selected, since it is between 6.7 and 9.75 gal.

Example 3: Select a compression tank for the system in *Example 1* if the system height is changed to 30 ft.

Solution:

(*a*) From Table 1 it is found that an initial pressure of 18 psig can be used at static heights up to 32 ft.

(*b*) From Table 1, the correction factors at 18 psig are:

$$\text{for maximum tank size, } 1.74;$$
$$\text{for minimum tank size, } 1.41.$$

(*c*) Maximum and minimum tank sizes will be:

$$\text{maximum tank size} = 13.9 \times 1.74 = 24.2;$$
$$\text{minimum tank size} = 13.9 \times 1.41 = 19.6.$$

(*d*) A 24 gal tank, capacity between 19.6 and 24.2, is selected.

Example 4: For the same conditions as *Example 3,* determine tank size if the tank is placed in the attic.

Solution:

(*a*) Establish the pressure rise. Initial pressure at boiler = 13 psig + 4 psig = 17 psig. Then the pressure rise = 30 psig (final pressure) − 17 psig = 13 psig.

(*b*) From the correction for attic tanks in Table 1, a pressure rise of 13 psig corresponds, by interpolation, to:

$$\text{maximum tank size correction factor} = \text{approx. } 0.61;$$
$$\text{minimum tank size correction factor} = \text{approx. } 0.475.$$

(*c*) The tank size is determined as:

$$\text{maximum tank size} = 13.9 \times 0.61 = 8.5 \text{ gal};$$
$$\text{minimum tank size} = 13.9 \times 0.475 = 6.6 \text{ gal}.$$

(*d*) An 8 gal tank, capacity between 8.5 and 6.6 gal, is selected.

Example 5: Select a compression tank for the system described in *Example 4* except that tank will be installed in the basement, the static head is 60 ft, and an air compressor will be used to charge the tank with air.

Solution:

(*a*) From the correction table for air compressor charged tanks, a static height of 60 ft requires an initial pressure on the system of 30 psig and a final pressure of 45 psig.

(*b*) Correction factors are then:

maximum tank size correction factor = 1.04;
minimum tank size correction factor = 0.82.

(*c*) Tank sizes will then be:

maximum tank size = $13.9 \times 1.04 = 14.5$;
minimum tank size = $13.9 \times 0.82 = 10.4$.

(*d*) A 12 gal tank is selected. This tank is much smaller than the 60 gal tank otherwise required for a basement tank under the conditions outlined.

Example 6: Select a compression tank for a panel or radiant heating system under the following conditions: 120,000 Btuh flash type boiler; 100,000 Btuh panel output; 12 psig initial pressure; 30 psig final pressure; 5 ft static head.

Solution: 12 psig initial, 30 psig final pressure and 5 ft static head will allow use of Table 1 without correction.

(*a*) From Table 1 the boiler and panel system capacities are:

120,000 Btuh flash boiler = 2.35 gal;
100,000 Btuh panel system = 1.75;
The maximum tank size = 4.1 gal;
The minimum tank size = $4.1 \times 0.73 = 3.0$ gal

(*b*) Assume that a tank between 3.0 and 4.1 gal is not available and that the next tank size is 8 gal. The requirement at 4.1 must be multiplied by 1.95 to equal 8 gal. The Table 1 initial pressure correction factor of 1.95 corresponds to 19 psig initial pressure.

The tank would then be sized at 8 gal with an initial pressure of 19 psig.

FORMULAS FOR DETERMINING TANK SIZE

Most of the analysis and other details necessary to develop the formulas are presented in Appendix B.

Factors which may affect selection of compression tank size are as follows:

1. Pressure volume or isothermal relationship of tank to system expansion.
2. Temperature changes of air within the tank as this affects pressure volume relationship.
3. Expansion of water in system.
4. Expansion of water in compression tank.
5. Expansion of piping, boiler and radiation.
6. Expansion of compression tank.
7. Liberation of air trapped in system water when heated.
8. The effect of trapping additional air in the compression tank during the filling of the system.
9. Loss of air from the compression tank.

NOMENCLATURE

p_a = atmospheric pressure, pounds per square inch, absolute.
p_f = initial fill pressure; pressure in tank under non-operating conditions, pounds per square inch, absolute.
p_o = final pressure; pressure in tank when operating, pounds per square inch, absolute.
p_e = additional pressure in tank over atmospheric, when water starts to enter during fill operation, pounds per square inch, gage.
p_L = limiting pressure existing at the highest point of the system under operating conditions, pounds per square inch, gage.
V_t = expansion tank size, gallons.
V_s = volume of system; boiler piping and radiation, gallons.
T_a = air temperature in tank before fill, Fahrenheit, absolute.
T_f = air temperature in tank after fill, Fahrenheit, absolute.
T_o = air temperature in tank during operation, Fahrenheit, absolute.
t_o = mean water temperature in system during operation, (or temperature −10 deg) required for boiling at limiting pressure p_L, Fahrenheit.
E_w = unit expansion of water.
E_p = unit expansion of piping, radiation and boiler.
E_t = unit expansion of compression tank.
E_{wt} = unit expansion of water contained in compression tank.
V_f = specific volume of saturated liquid.
α = coefficient of expansion.

If all factors involved are considered without evaluation, the general formula for determining compression tank size is

$$V_t = \frac{(E_w - E_p)V_s}{\frac{(p_a + p_e)T_f}{p_f T_a} - \frac{(p_a + p_e)T_o}{p_o T_a} - \left[1 - \frac{(p_a + p_e)T_f}{p_f T_a}\right] E_{wt} + E_t} - 0.02V_s \quad (1)$$

The symbols in Equation 1 are defined in the Nomenclature.

Equation 1 is rather complex, and is not necessary for the selection of a compression tank for the average system (not including the safety loop system) for which the following equation is satisfactory:

$$V_t = \frac{(E_w - E_p)\,V_s}{\frac{p_a}{p_f} - \frac{p_a}{p_o}} - 0.02V_s \quad (2)$$

Equation 2 is the result of dropping many of the factors of small importance from Equation 1.

Equation 2 and the formula given on p. 525 of THE GUIDE 1952 are similar.

Boundary limits for compression tank sizing are:

1. *(Upper)* Boiling must not occur in any part of the system at any temperature-pressure relationship in the design range.

2. *(Lower)* Relief valve operation should not occur at temperatures in the design range.

The lower boundary limit can be evaluated from Equation 2 by substituting values of E_w and E_p corresponding to the filling and operating temperatures,

respectively, in Equation 2. The reference temperature, from which expansion of the system will be taken, is established at 40 F. In some systems the reference or fill temperature may be higher than that established, but minor changes at the lower temperature level will not affect the final result appreciably.

Unit expansion of water (E_w) has been determined from the relationship

$$E_w = (V_f @ t_o)/(V_f @ 40\ \text{F}) - 1$$

Results are plotted in Fig. 1.

Unit expansion of pipe (E_p) is determined from the relationship $E_p = 3\alpha/\text{F}$, where $\alpha = 6.85 \times 10^{-6}$, the coefficient of expansion for wrought iron. Results are plotted in Fig. 2.

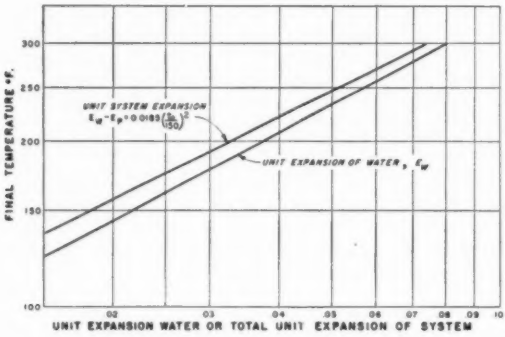


FIG. 1. UNIT EXPANSION OF WATER OR TOTAL UNIT EXPANSION OF SYSTEM (INITIAL TEMPERATURE 40F)

The two curves thus established have been combined in a final curve of $E_w - E_p$, also plotted in Fig. 1. From this curve the following formula was developed to express unit system expansion in terms of mean system temperature:

$$E_w - E_p = [0.0185\ (t_o/150)^2] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The upper boundary limit concerns boiling in the system. It may be evaluated from Equation 2 by substituting values of E_w and E_p corresponding to the temperature and pressure existing at the point of lowest pressure (the highest point) in the system.

Fig. 3, a log-log plot of boiling temperature *vs.* pressure, indicates the temperature-pressure relationship which must not be exceeded. Parallel to a line passing through the 10 psig and 30 psig points and at a temperature 10 deg lower than the values given in the steam tables, a straight line curve has been established. This curve will be used as the upper mean temperature-pressure relationship which must not be exceeded and is based on accepted 20 deg temperature drop design practice; the maximum temperature reached in the system is 10 deg above the mean temperature upon which calculations must be based.

As higher pressure and temperature relationships are reached, mean temperature deviates more and more from the actual boiling conditions. In this paper, the upper limit of usefulness of the curve is 275 F. The curve can be represented by the equation:

$$t_o = 170\ p_L^{0.1295}$$

For high temperature systems a new curve must be established.

Substituting for t_o in Equation 3, yields

$$E_w - E_p = 0.0236\ p_L^{0.259} \quad \ldots\ldots\ldots\ldots \quad (4)$$

Equation 4 expresses unit system expansion in terms of a limiting pressure related to a temperature which is 10 deg below the boiling point of water at that limiting pressure.

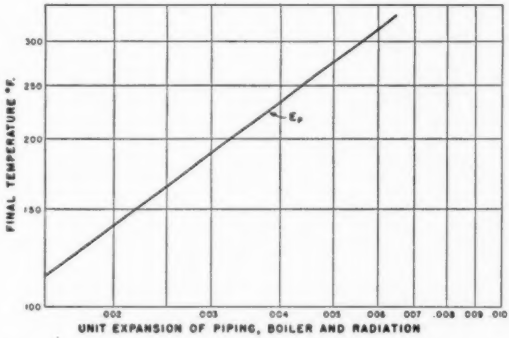


FIG. 2. UNIT EXPANSION OF PIPING, BOILER AND RADIATION (INITIAL TEMPERATURE 40 F)

Substituting values of $E_w - E_p$ from Equations 3 and 4 in Equation 2, two new equations are obtained to express the boundary limitations existing for sizing of compression tanks.

$$V_{t,max} = \frac{0.0236\ p_L^{0.259} V_s}{\frac{p_a}{p_f} - \frac{p_a}{p_o}} - 0.02 V_s \quad \ldots\ldots\ldots \quad (5)$$

$$V_{t,min} = \frac{0.0185 \left(\frac{t_o}{150}\right)^2 V_s}{\frac{p_a}{p_f} - \frac{p_a}{p_o}} - 0.02 V_s \quad \ldots\ldots\ldots \quad (6)$$

A limitation of these two equations which does not appear as a term is the condition that air must not escape from the tank if waterlogging of the tank is to be avoided.

In Equation 5 the term $V_{t,max}$ is the compression tank size required under maximum temperature-pressure conditions.

In Equation 6 the term $V_{t,min}$ is the compression tank size required under minimum temperature-pressure conditions.

The term p_a is the pressure within the tank before water enters. Normally this pressure is considered atmospheric pressure. This premise, however, is based on slow filling of the system and on venting when filling. If the system is not vented, or if the filling rate is over approximately 4 gpm (approximate automatic fill valve rate) the tank pressure will be above atmospheric before water starts to enter.

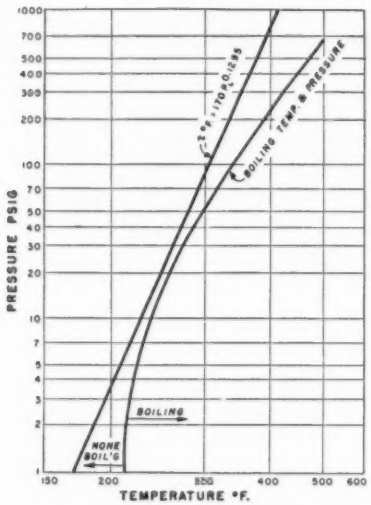


FIG. 3. TEMPERATURE-PRESSURE RELATIONSHIP FOR BOILING WATER

At a 10 gpm filling rate, pressure in the tank will be approximately 1 psig in a system with flash-type boiler and radiation above the main; about 3½ psig in a tank with conventional-type boiler and radiation above the main and approximately 7 psig in a tank where the boiler is above the radiation. An increase of p_a above atmospheric pressure decreases tank size required but is difficult to control.

The term p_f in Equations 5 and 6 expresses the pressure in the tank after filling while the system is still cold. Normally 12 psig, it may be either larger or small depending on the static height of the system above the tank. Minimum usable pressure for p_f may be considered as the static height of the system above the tank plus 4 psig, where 4 psig is the pressure required at the top of the system.

Raising p_f will increase the tank size required and vice versa. On large systems, where a small tank is decidedly advantageous, the tank is often installed at as high a point as possible, in order to reduce p_f and, hence, the tank size.

The term p_o, the final desired operating pressure at the tank, may cover a wide range. Normally 30 psig, it may be higher or lower depending on static height above the tank or desired operating conditions. With a given p_f, a larger p_o reduces the tank size required.

The term p_L expresses the highest pressure which may exist at the highest point of the system under desired maximum operating conditions. A higher p_L establishes a higher maximum final operating temperature and vice-versa.

For any specific system, p_L can be determined by adding filling pressure at the highest part of the system to desired pressure rise in the system:

$$p_L = 4 \text{ psig} + (p_o - p_f),$$

where 4 psig is minimum desired filling pressure at highest point of the system. This pressure may be set higher at the discretion of the designer.

In Equation 6, the term t_o, the temperature at which relief pressure will occur, does not account for the possibility of boiling. The term is used only as a lower temperature compression tank sizing limit.

The term $0.02\ V_s$ refers to the effect of additional air being brought into the tank through freeing the water of its absorbed air.

The term V_s refers to the amount of water contained in the system.

CONCLUSIONS

The effects of vapor pressure within the tank, as this affects tank sizing, should be examined. The effect of vapor pressure for the great majority of systems, however, will not be of great importance since the tank temperatures are not high. The effect of vapor pressure may be more significant in a safety loop system.

The tank sizing method developed is simple and, based on comparison with existing methods, gives a close approximation of tank requirements.

Actual test work and application data would aid in determining the accuracy of results obtained from Table 1 and the equations.

APPENDIX A

SOURCE OF VOLUME DATA

Fig. A-1 is drawn from data received from boiler manufacturers giving water content for specific boilers of both conventional and flash types. The water content is plotted against the net Btu rating of the boiler. The points fall in a fairly straight line, although there is a spread for boilers at the same rating. Average water content for the boilers has been established at the low water content side of the curve in order to insure against boiling through over-sizing the compression tank with respect to the system as might occur by basing calculations on a high-water-content boiler and then actually using a low-water-content boiler.

Fig. A-2 shows the water content for a number of heating systems; one-pipe, two-pipe and panel. Average water content is again established at the lower water content side of the curve.

Average values for radiation water content have been obtained from manufacturers with the following results:

Large tube radiation = 0.114 gal/sq ft water radiation;
Thin tube radiation = 0.056 gal/sq ft water radiation;
Convectors = 0.025 gal/sq ft water radiation;
Baseboard = 0.013 gal/200 Btuh output.

OPERATION OF PRESENT DAY SYSTEMS

Most hot water radiation systems are operated under the following conditions:

p_a = atmospheric pressure.
p_f = initial or fill pressure = 12 psig.
p_o = final or operating pressure = 30 psig.
p_L = limiting pressure at top of system; maximum tank sizing condition = 4 + (30 − 12) = 22 psig.
$t_{o,\,max}$ = limiting temperature corresponding to 22 psig; maximum temperature at which system reaches relief pressure = 262 F.
$t_{o,\,min}$ = lower temperature limit at which system reaches relief pressure = 220 F

These conditions are considered as standard in this paper.

By applying the conditions given to Equations 5 and 6, it appears reasonable that,

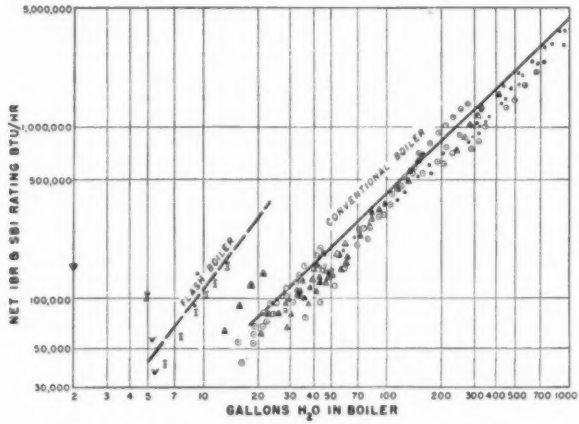


FIG. A-1. BOILER WATER CONTENT IN GALLONS

under standard conditions, the compression tank should be sized between 16 and 22 percent of the total system volume.

Panel systems differ from the majority of hot water heating systems in that the maximum panel temperature rarely exceeds 140 F. Since boiling in the system need not be feared, the minimum tank size can be established by Equation 6, using a temperature t_o of 140 F and maintaining other conditions as standard. Calculations indicate that the panel system requires only 5.3 percent of the system volume for minimum compression tank size. Maintaining the same maximum and minimum ratio as for the standard radiation system, the maximum tank size required would be 7.3 percent, with a corresponding maximum temperature of 160 F. An all panel system does not always require extremely accurate tank sizing, and the limits indicated may be increased somewhat. Care should be exercised, however, for systems containing panels and radiators, and systems in which a high water content boiler is maintained at high temperature.

BASIS FOR CORRECTION FACTORS IN TABLE 1

When the maximum tank size required for standard conditions has been selected by reference to Table 1, correct for conditions other than standard by referring to the

correction factors of Table 1. The conditions for which corrections may be made are given in the following items:

1. *Correction factors for change in initial pressure when final pressure is maintained at 30 psig.* As has been previously noted, a higher initial pressure requires a larger tank and vice-versa. Table 1 lists correction factors for various initial pressures other than 12 psig together with the maximum applicable static height of the system. The table should prove useful for high static head systems and for large low static head systems in which a reduction of tank size is desired.

2. *Correction factors for attic tanks.* In order to reduce initial tank pressure and thus reduce size, the compression tanks on many high static or very large systems

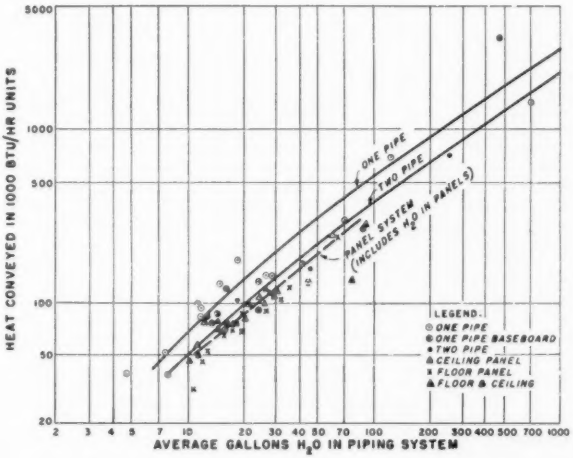


FIG. A-2. GALLONS OF WATER IN PIPING SYSTEM

are installed at the highest point of the system. Because the relief valve setting on high static head systems is not a constant, varying from 30 psig to 160 psig plus, the correction table is based on pressure rise of the system from its cold to operating condition. Pressure rise is defined as final boiler pressure minus initial boiler pressure, initial boiler pressure being considered as system static head + 4 psig.

3. *Correction factors for safety loop sizing.* In a small minority of systems, in which the main is run below the top of the boiler, the compression tank is used secondarily as a heat dissipating device under runaway boiler heat input conditions. This system differs from most in that gravity flow to the tank is encouraged by using a recirculating leg connected from the tank to the bottom of the boiler. The recirculating leg is in addition to the normal boiler to tank connection used on all compression tank systems. Because the temperature in the tank is higher in the safety loop system than in the average system, Equation 2 of the paper is modified as follows:

$$V_t = \frac{(E_a - E_c)V_s}{\frac{p_a}{p_f} - \frac{p_a T_f}{p_o T_a} - \left|1 - \frac{p_a}{p_f}\right| E_{mf} + E_f} - 0.02 V_a \qquad \text{(A-1)}$$

Since the safety loop system is used only when all radiation is below the boiler, high initial pressures in the tank are unnecessary, and the correction table has been made using only low initial fill pressure from 4 psig to 12 psig.

4. *Correction factors for systems using compressed air to charge the tank.* On large systems in which an attic tank cannot be used it is economical to use compressed air for charging basement tanks. A smaller tank will then be required because almost the entire tank can be considered as an air chamber. In this paper, the tank is considered 80 percent full of air under initial or cold conditions. Under these conditions, the equation on which the correction table is based will be as follows:

$$V_t = \frac{(E_w - E_p)V_s}{0.8\left(1 - \frac{p_a}{p_f}\right)} - 0.025V_s \quad \text{(A-2)}$$

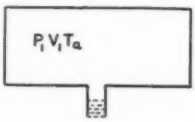
5. *Correction factors for installed tanks which are undersized.* Under unusual conditions, when the amount of water in a specific system is much greater than the average amount on which Table 1 is based, the system will reach relief pressure at low temperature. By the correction factors in Table 1, the necessary increase in *effective tank size* can be determined for the temperature at which the relief valve operates. The *effective tank size* may be increased by lowering the initial pressure, changing the location of the tank, or increasing its size.

APPENDIX B

A. DEVELOPMENT OF GENERAL EQUATION FOR COMPRESSION TANKS CONSIDERING ALL FACTORS INVOLVED

In order to develop the general equation for compression tanks a typical compression tank will be considered from its empty to full operating condition. For definitions of symbols see Nomenclature.

(*a*) *Empty:*

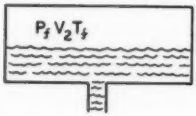


where

$p_1 = p_a + p_o$
$V_1 = V_t + 0.02\ V_s$
$0.02V_s$ = liberated air from water in system.

Considerations: Water is about to enter tank. In addition to atmospheric air some additional air may have been forced into the tank when filling, thereby increasing the pressure above atmospheric. Additional atmospheric air liberated from the water after heating may also reach the tank.

(*b*) *After initial fill:*

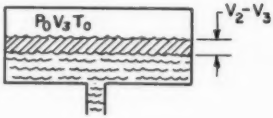


where

$$V_2 = (p_1 V_1 T_1)/(p_f T_a)$$

Considerations: Water has now entered tank changing pressure, temperature and volume of air.

(*c*) *During operation:*



where

V_3 = final air volume.

Considerations: Heating of the system has increased the volume of water in the tank and the system piping, resulting in a total increase of water in the tank of $V_2 - V_3$. A relation between initial air volume in tank and $V_2 - V_3$ in terms of pressure and temperature is needed as follows:

$$V_3 - V_3 = V_1 \left(\frac{p_1 T_f}{p_f T_a} - \frac{p_1 T_0}{p_0 T_a} \right) \quad \ldots\ldots\ldots\ldots \text{(B-1)}$$

$V_2 - V_3$ represents increase in total system volume relative to the tank. This can be represented as:

$$V_2 - V_3 = V_3 (E_w - E_p) + V_{wt} E_{wt} - V_t E_t$$

V_{wt}, the volume of water in tank after fill $= V_1 - V_2$ and $V_2 = V_1 p_1 T_t / p_t T_a$. The following equations can be made much simpler and more practical if V_1 is substituted for V_t in the term $V_t E_t$. A slight theoretical error occurs in the final expression which for practical purposes is negligible. Substituting for $V_2 - V_3$ in Equation B-1 yields:

$$V_3 (E_w - E_p) + V_1 \left(1 - \frac{p_1 T_f}{p_f T_a} \right) E_{wf} - V_1 E_f =$$

$$V_1 \left(\frac{p_1 T_f}{p_f T_a} - \frac{p_1 T_0}{p_0 T_a} \right)$$

Solving for V_1

$$V_1 = \frac{(E_w - E_p) V_3}{\dfrac{p_1 T_f}{p_f T_a} - \dfrac{p_1 T_0}{p_0 T_a} - \left(1 - \dfrac{p_1 T_f}{p_f T_a} \right) E_{wf} + E_f}$$

By substituting $V_t + 0.02V_s$ for V_1 and $p_a + p_e$ for p_1, the equation becomes:

$$V_t = \frac{(E_w - E_p)V_s}{\frac{(p_a+p_e)T_f}{p_f T_a} - \frac{(p_a+p_e)T_o}{p_o T_a} - \left[1 - \frac{(p_a+p_e)T_f}{p_f T_a}\right] E_{wt} + E_t} - 0.02V_s \quad \text{(B-2)}$$

B. Evaluation of General Formula Considering Effect of Factors

(*a*) *Pressure-volume or isothermal relationship of tank to system expansion:* The factor simply involves the application of the gas laws to the pressure volume relationship.

(*b*) *Temperature changes of air within the tank as this affects pressure volume relationship:* Temperature T_a, air temperature in tank before filling, will vary between limits of (460 + 40) F abs to (460 + 90) F abs, depending on the time of year the system is filled. Temperature T_f, would at the instant the system reaches 12 psig, be equal to temperature T_a. With filling water at 40 F there might be a period of time during which air temperature T_f would drop below T_a. The gas law relationship would still hold, so that a change in T_f would not change the basic equilibrium but would only vary p_f and V_2 directly with the change in temperature. From the foregoing $T_a = T_f = 70$ F, approximately. Temperature T_o remains to be evaluated. The temperature of the air in the tank would vary directly with the amount of gravity flow permitted. On most systems, and especially those having a device to limit gravity flow, the air temperature in the tank would approximate basement air, plus a small temperature rise depending on gravity flow. (460 + 70) F, abs $< T_o <$ (460 + 90) F, abs. The safety loop system would probably have an air temperature approaching boiler water temperature. Leaving the analysis of the safety loop system for later consideration, the general equation can be reduced to:

$$V_t = \frac{(E_w - E_p)V_s}{\frac{p_a+p_e}{p_f} - \frac{(p_a+p_e)T_o}{p_o T_a} - \left[1 - \frac{p_a+p_e}{p_f}\right] E_{wt} + E_t} - 0.02V_s \quad \text{(B-3)}$$

(*c*) *Expansion of water in system:* This factor is of great importance and must be left in the equation.

(*d*) *Expansion of water in compression tank:* Usually, compression tanks do not reach temperatures high enough to warrant consideration of the water in the tank except in the case of the safety loop system. The term E_{wt} can therefore be dropped from the expression for the average system.

(*e*) *Expansion of compression tank:* Following the same reasoning as in item (*d*) the term E_t can be dropped for the average system.

(*f*) *Expansion of piping, boiler and radiation of system:* The volume of water expansion alone does not give a true picture of total system expansion for, as the water expands, the piping, boiler and radiation must also expand giving a total expansion, relative to the system, of water expansion minus system expansion. The term E_p must remain in the expression.

Considering factors from (*d*) to (*f*) the general equation reduces to:

$$V_t = \frac{(E_w - E_p)V_s}{\frac{p_a+p_e}{p_f} - \frac{(p_a+p_e)T_o}{p_o T_a}} - 0.02V_s \quad \text{(B-4)}$$

(*g*) *Liberation of dissolved air when heated:* Water at 40 F will contain about 0.024 gal of dry air per gallon of water and at 180 F will contain about 0.004 gal of dry air per gallon of water. Thus if one gallon of water is heated from 40 F to 180 F about 0.02 gal of dry air will be liberated. The volume of liberated air in system water is provided for in the equation by 0.02 V_s.

(*h*) *Consideration of waterlogging of compression tank:* Systems employing a relatively large and direct pipe connection from the boiler to the compression tank will waterlog after a period of time. Waterlogging of the compression tank will invalidate any method of tank sizing since, after a time, the amount of air in the tank is undeterminable.

(*i*) *The effect of trapping additional air in the compression tank during filling of the system:* The problem basically is concerned with the rate of filling with respect to the air venting capacity of the system. The method of filling should be examined. If the system is vented after initial fill pressure is reached, the tank will have a maximum amount of additional air. If venting is carried on during the fill operation the tank will have a minimum amount. The assumption can be made that most systems will be vented during fill since some time is saved in using this method.

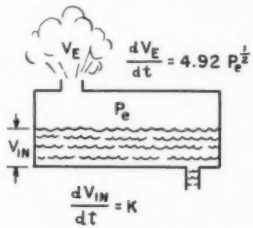
The nozzle diameter available for venting can be approximated at 1/16 in. diameter An approximation for the discharge of free air through a 1/16 in. nozzle can be given as:

$$dV_E/dt = 4.92\ p_e^{1/2}$$

where

p_e = pressure causing flow in pounds per square inch gage.
dV_E/dt = flow of free air, gallons per minute.

Filling rates can be established as varying between 1 to 20 gpm. Automatic fill valves are installed on about 60 percent of all systems today. They restrict the filling rate to between 3 and 4 gpm.



For a tank from which air is escaping at a rate depending on internal pressure and into which water flows at a rate K the following equation can be written:

$$p_a(V_t - V_E) = (p_e + p_a)(V_t - V_{IN})$$

where

p_a = atmospheric pressure, pounds per square inch, absolute.
p_e = pressure existing in tank at any time when V_E air has escaped and V_{IN} in water has entered pounds per square inch, gage.
V_t = total volume of tank.
V_{IN} = water volume in tank at any time, gallons, or volume of water in system when water starts to enter tank during fill, gallons.

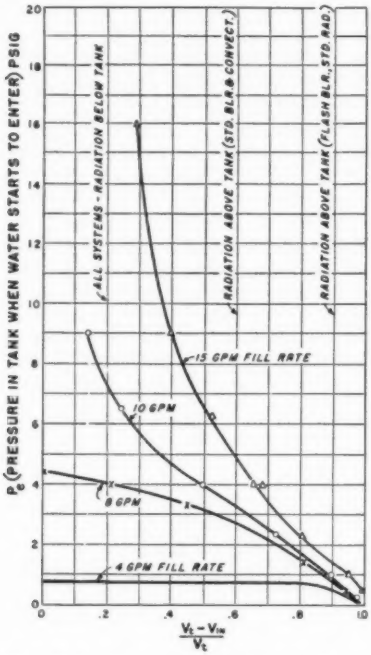

P_e (PRESSURE IN TANK WHEN WATER STARTS TO ENTER) PSIG
ALL SYSTEMS - RADIATION BELOW TANK
RADIATION ABOVE TANK (STD. BLR. & CONVECT.)
RADIATION ABOVE TANK (FLASH BLR., STD. RAD.)
15 GPM FILL RATE
10 GPM
8 GPM
4 GPM FILL RATE
$\frac{V_t - V_{IN}}{V_t}$

FIG. B-1. COMPRESSION TANK PRESSURE RISE DURING FILLING

The equation can be differentiated with respect to time and, after proper substitution for filling and air escape rates, can be integrated between proper limits of time and pressure to obtain the following equations:

For filling rates less than 9½ gpm

$$\frac{V_t - V_{IN}}{V_t} = \frac{\left[\frac{2K\sqrt{p_e}\left(-72.2-\sqrt{5220-58.8K^2}\right)+58.8K^2}{2K\sqrt{p_e}\left(-72.2+\sqrt{5220-58.8K^2}\right)+58.8K^2}\right]^{\frac{72.2}{\sqrt{5220-58.8K^2}}}}{\frac{p_e}{14.7}-\frac{4.92\sqrt{p_e}}{K}+1} \quad \text{(B-5)}$$

For filling rates greater than 9½ gpm

$$\frac{V_t - V_{IN}}{V_t} = \frac{1}{\left(\frac{p_e}{14.7}-\frac{4.92\sqrt{p_e}}{K}+1\right)e^{\frac{144.4}{\sqrt{58.8K^2-5220}}\left[\tan^{-1}\frac{2K\sqrt{p_e}-72.2}{\sqrt{58.8K^2-5220}}-\tan^{-1}\frac{72.2}{\sqrt{58.8K^2-5220}}\right]}} \quad \text{(B-6)}$$

where

K = filling rate, gallons per minute.

The complete development of the equations may be obtained from the authors.

Fig. B-1 shows the pressure rise in the tank when water starts to enter for the condition that air vents through one 1/16 in. diameter orifice at an air flow rate of 4.92 $p_s^{1/2}$.

As will be noted, the pressure rise at 4 gpm fill rate approximates 0.8 psig rise for any type system. The pressure rise for higher fill rates can be substantially above this value, varying for the type of system.

Most hot water heating systems use pressure reducing values for filling purposes, restricting fill flows to 3 to 4 gpm with the result that pressure in the tank when water starts to enter is probably around 0.8 psig. Since the tank sizing method must be based on average system p_s will be considered as equal to 0.8 psig, though the fact that pressure rise at higher fill rates than those described can be considerably higher, should be recognized.

The general equation can then be stated as:

$$V_t = \frac{(E_w - E_p)V_s}{\frac{p_a + 0.8}{p_f} - \frac{(p_a + 0.8)T_o}{p_o T_a}} - 0.02V_s \quad \text{(B-7)}$$

Utilizing the approximate temperatures for T_o and T_a as given in item (*b*) and under standard conditions, it can be determined that the effect of temperature and of $p_s = 0.8$ psig are such that they approximately counterbalance one another.

The equation, dropping both p_s and temperature considerations, then becomes:

$$V_t = \frac{(E_w - E_p)V_s}{\frac{p_a}{p_f} - \frac{p_f}{p_o}} - 0.02V_s \quad \text{(B-8)}$$

which is Equation 2 of the paper.

C. DEVELOPMENT OF EQUATION FOR THE SAFETY LOOP SYSTEM

(*a*) The safety loop system differs from the average system in that the temperature of the tank approximates boiler temperature.

(*b*) Evaluation of factors generally follows those for the average system except that expansion of water in the compression tank and expansion of the tank itself must be considered. The general equation then reduces to:

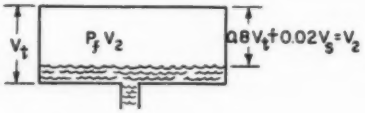
$$V_t = \frac{(E_w - E_p)V_s}{\frac{p_a}{p_f} - \frac{p_a T_o}{p_o T_a} - \left[1 - \frac{p_a}{p_f}\right] E_{wt} + E_t} - 0.02V_s \quad \text{(B-9)}$$

Consideration of the fact that air temperature in the tank will be less than Boiler Temperature allows substitution of p_a for term $(p_a + 0.8)$.

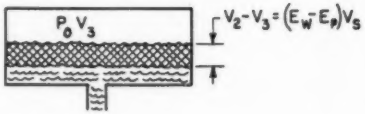
D. DEVELOPMENT OF EQUATION FOR SYSTEM USING COMPRESSOR TO CHARGE TANK WITH AIR

Assume tank 0.8 full of air after initial pressure has been reached. Additional air may be added later after liberation of entrapped air in system.

(*a*) After fill:



(*b*) During operation:



$$V_2-\frac{P_f V_2}{P_o}=(E_w-E_p)V_s;\ V_2=\frac{(E_w-E_p)V_s}{1-\frac{P_f}{P_o}}$$

Substituting: $0.8V_t+0.02V_s=V_2$ *yields*:

$$V_t=\frac{(E_w-E_p)V_s}{0.8\left(1-\frac{P_f}{P_o}\right)}-0.025V_s \quad \text{(B-10)}$$

Which is Equation A-2 of Appendix A.

DISCUSSION

N. D. SKINNER, Indianapolis, Ind. (WRITTEN): The authors of this paper have approached this rather difficult problem in a very practical and helpful manner. In preparing data for the selection of compression tanks, careful consideration has been given to the type of boiler and radiation used, the nature of the piping system, the necessary temperature correction, whether the tank is to be attic or pressure type, and special consideration is given to radiant panel systems.

The material contained in Table 1 seems to cover the aforementioned system variations sufficiently to give information necessary to making a precise selection of a compression tank for a hot water system. The problem is a common one and it has been perplexing our industry for many years, inasmuch as various methods in practice have produced results over a rather wide range. With more accurate information available, it will be possible for designers of hot water heating systems to select compression tanks with a greater degree of assurance and accuracy in the future.

This paper also includes development of an equation for selecting a compression tank for a system where an air compressor might be necessary to automatically maintain the proper amount of air at all times. Experience in the past indicates that, in many cases, compression tanks have been selected which were larger than required, and resulted in the waste of material. Standardization of the procedure of selection of these tanks in the future will, undoubtedly, reduce cost, and provide for greater accuracy.

R. C. CHEWNING, Portland, Ore.: The authors' method of estimating tank sizes fills a very definite need in the work of engineers designing hot water systems. The

method of tabulating water content according to system components facilitates the use of the suggested expansion tank sizing method discussed in this paper.

As stated by Mr. Lockhart, there is still more work to be done on the suggested method. A study of Fig. A-2, which shows gallons of water in piping system, reveals only four systems plotted with capacities greater than 400 Mbh. Use of the method in sizing expansion tanks for larger size installations, necessitates more data on the water content of larger installations to insure that the proper size expansion tank is provided.

A number of things in this paper are a little new in approach. The fact that the amount of expansion in the system was calculated on the basis of a final temperature of 262 F is unusual. A normal system runs at a maximum temperature of approximately 210 F. High-temperature systems are coming into use but, in establishing 262 F as the temperature for calculating expansion, and then using the water volumes for the various components of the systems on the low-volume side of the curve developed rather than on the high-volume side, complications are added in addition to inaccuracies in the engineering calculations. In effect, one factor tends to cancel out the other. Use of the high temperature increases the size of the expansion tank, while estimating a lower volume of water for the system decreases the size of the expansion tank.

This paper presents a method and a tool to engineers which will be of great help to them and the future additional work will make it even more helpful.

M. C. CHRISTESEN, Newark, N. J.: Did the authors include commercial boilers or so-called generators or packaged units in the conventional boilers? Is there any difference due to firing rate?

Do these few figures conform to the *A.S.M.E.* requirements?

JOHN W. JAMES, Chicago, Ill.: Because of the trend in recent years, toward incorporating the compression tank in the upper part of a so-called packaged heating boiler, it seems sensible to learn from the authors whether the method of selecting tank sizes discussed in this paper can be used with packaged boiler designs.

Reference has been made to the inclusion of material on the selection of expansion tanks in Par. H-39 and H-92 of the 1949 edition of the *A.S.M.E.* Low Pressure Heating Boiler Code. It is recognized that in this code selection procedure is based on gravity hot water heating systems. It is hoped, that from the presentation of data in this paper, recommendations can be prepared for possible revision of the existing *A.S.M.E.* rules covering this important subject.

Another important aspect of this problem has to do with proper location of the compression tanks to achieve desirable system operating characteristics. It is true that a great number of compression tanks are located in the boiler room, but it is also recognized that there are a number of designers who prefer to locate the tanks at some high point in the system. A few years ago, the National Board of Boiler and Pressure Vessel Inspectors conducted an investigation on the proper location of a compression tank in a system. These results were reported in a paper* entitled, *Tests of Hot Water Pressure Relief Devices,* by Prof. Paul Bucher.

The tendency toward waterlogging of compression tanks located in the boiler room of a heating system has been greatly reduced in recent years by the use of proprietary devices, but there are still a number of installations in which the problem of air transfer from the compression tanks to some higher point in the system presents a real air elimination problem.

AUTHORS' CLOSURE (G. F. Carlson): Mr. Chewning's comments regarding the necessity for additional work are pertinent. It is believed, however, that the curves

* (Proceedings of the 18th General Meeting of the National Board of Boiler and Pressure Vessel Inspectors, Columbus, Ohio, 1948.)

on system water content are good averages, since they have checked with installed systems.

Use of 262 F as the temperature for calculating expansion and then using system water volumes on the low water content size can be justified. In order that the described method of sizing compression tanks be practical, as wide a spread of satisfactory tank sizes as possible should be provided within safe limits. This requirement, plus the fact that many systems now operate in the 250 F range, necessitated consideration of the high temperature limit at as high a point as possible, considering a 30 psig final pressure.

At the same time, care had to be taken that pressure temperature limitations at 30 psig would not, under any circumstances, be exceeded. This possibility is guarded against by utilizing water contents on the low volume side. While it seems that one factor tends to cancel the other, it is the only method by which requirements of both practicability and safety can be met.

Concerning Mr. Christesen's questions, commercial boilers and generators were included and there would be no difference in compression tank size due to firing rate.

Whether the figures conform to *A.S.M.E.* requirements is somewhat difficult to answer since *A.S.M.E.* is now in the process of revising its requirements.

In regard to Mr. James' comments, pending further study of the effects of vapor pressure within the tank on tank sizing, we would not recommend the method described in this paper for use in sizing compression tanks for packaged boilers.

The comments regarding waterlogging of compression tanks are very important since waterlogging of the compression tank will invalidate any method of tank sizing.

The use of a dip tube fitting at the boiler will reduce the possibility of waterlogging for systems with compression tanks located at the boiler room.

1466

No. 1466

THE BACTERIA AND ODOR CONTROL PROBLEM IN OCCUPIED SPACES

By R. L. KUEHNER*, YORK, PA.

CONSISTENT PROGRESS toward the solution of the problems of air sanitation has been confounded by recent frantic, and in many cases, haphazard efforts to satisfy the consuming public. This is illustrated by the number of products parading under the badge of bacteria and odor *removers.* Extensive experimental work has been undertaken toward an understanding of the basic problems of air sanitation. The results permit the interim formulation of recommendations concerning bacteria and odor control which will better direct the efforts of the researcher and permit the understanding of the layman.

This discussion applies to the absolute removal and destruction of bacteria and odor, not to the masking or neutralizing of odors or reducing the sensitivity of the olfactory sense.

INTRODUCTION

The term, *air sanitation,* is borrowed from Dalla Valle[1]. While his interpretation appears to limit its meaning primarily to dust control, and secondarily to bacteria and odor control, this term in its liberal sense should be a basic division identified with air conditioning. Air conditioning has meant, too often, simple cooling and dehumidification of air. The future demands that this industry assume the responsibility for air conditioning in its broadest sense. How much this will entail only can be guessed, since there is no full understanding of the finite differences between *country fresh* and *vitiated stale* air. By concerted effort, the standards of temperature, relative humidity, and air motion have been set and become the basis of air conditioning practice. By the same concerted effort, sanitation qualities must be standardized before claim can be

* Research Engineer, York Corp.

[1] Exponent numerals refer to Bibliography.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

made to truly conditioned air. Whether these qualities are the poisons in expired breath mentioned by Hammond[2], the anaphylactic protein of Rosenau and Amoss[3], the ions of Dessauer[4] and Yaglou[5], or as yet unknown factors, the production of truly conditioned air is the obligation of this industry to society. A man consumes daily, pound for pound, several times more air than food, and 10 times as much air as water[6]. Does not the air man breathes deserve the same quality control as his food and water?

Over and beyond this altruistic obligation is the practical side. Until a satisfactory air sanitation means is found, the industry is faced with the penalty of outside air required to preserve decent indoor conditions. U. S. Naval research[7] has established that only one cfm outside air per person is necessary to maintain proper oxygen and CO_2 levels, yet 20-30 cfm outside air per person is common practice for ventilation purposes. Consider the reduction in first cost and operating cost when an air sanitizer is substituted for outside air. While research is progressing to define the suspected components and discover unknown components of air quality, it is only in bacteria and odor control that the problems are sufficiently recognized, and progress sufficiently great, to make a real contribution in this field.

It is beyond the scope of this paper to trace the history of bacteria and odor control. Nor is it intended to assay the worth, or lack thereof, of literature in this field. Interest in air-borne bacteria dates from the work of Louis Pasteur around 1860. Interest in odor dates from man's creation. Odors and Sense of Smell[8] cites references back to 320 B.C. in a 300-page bibliography. The thousands of pieces of literature are more descriptively presented than could be done here in such texts as those by Moulton[9], McCord[10], Medical Research Council[11], Rosebury[12], Crocker[13], and Moncrieff[14].

Yet with all this interest and knowledge there is no universally accepted air sanitizing agent. A great handicap, recognized by Gerhardt[15] in 1928 and by Crocker[16] in 1935, has been the lack of a common working language. This is only a symptom. There has been no complete study of the nature, cause and effect of bacteria and odor in enclosed spaces. For example, it is recognized by all that the spread of infection via the air is potentially a great threat, increasing as more and more people are brought in close contact and the same air is delivered from person to person. Since no one can say at what point air is safe or harmless, the penalty of outside air must be continued because there is no yardstick to measure a substitute method. Odor, while not physiologically harmful, is more tangibly objectionable. Because their nature, their rate of production, their maximum tolerable levels and necessary rates of removal are unknown, there is no satisfactory progress toward their control.

It was with the goal of developing an acceptable baseline of understanding that this work was undertaken. In so doing it was necessary to establish, in a broad sense, those air components concerned and to define the degree of removal of these components required. This obviously demands methods of measuring these components, not in past terms of good or bad, pleasant or unpleasant, but in finite numbers. It demands the removal of opinion. Only by so doing can the researcher evaluate his progress, the producer know what he is selling and the consumer know what he is buying. Odor and bacteria control has become a highly competitive field but one in which the competition remains in the realm of adjectives to such an extent that the air conditioning industry

could not now by any single criterion, or group of criteria, determine which deodorant or disinfectant is better than the other, or if any have merit at all.

ODOR

Experimental Procedure: In 1946[17] literature was presented to establish that odors are specific chemical substances of fat soluble and certain minimum vapor pressure characteristics. The first conclusion drawn is the finite nature and indefinite effects of odors. While it is possible to find and identify a specific chemical substance of certain consistent physical characteristics at the root of every odor, it has been impossible to conclude why the sense of smell is stimulated and, accordingly, impossible to formulate any generalization as to odor types and methods for their destruction.

If odors were of a specific chemical type and their behavior patterns followed this chemical type, it would be simple to undertake odor problems on the basis of individual odorous chemicals. However, investigation has shown that because a given chemical or mechanism is deodorant toward any single odor, it need not be deodorant toward naturally occurring odors which are aggregates of many individual odors. In the laboratory, a deodorant was developed using butyl mercaptan as the test odor. When this deodorant was checked in a practical system with tobacco smoke, without fresh air, it was found that while the mercaptans present in tobacco disappeared the system became foul because of the build up of the other odorous components which the deodorant would not attack. The deodorant was specific for mercaptans. Because the same specificity occurs if one uses odorous acids, bases, alcohols, aldehydes or others as test odors it becomes impossible to use any synthetic odor as characteristic of naturally occurring odors except in those rare cases, such as egg rooms, where one specific odor is the offender.

Therefore, any test work must be done on naturally occurring odors. Tobacco odor and body odor were considered the most troublesome in air conditioning. Because both of these are aggregates of many odors, unknown to a great extent, a chemical means of measuring their concentration seemed remote.

Dilution Measurement: Since all work on the removal of odors must, in the final analysis be put in terms of human sensibilities, attention was directed toward a *dilution-threshold* method. Because the olfactory threshold is the minimum concentration of an odor which can be smelled and because increases in concentration above this minimum cannot be evaluated by the unaided nose, this technique reduces all odorous air to threshold values. By mixing sufficient purified air to a measured quantity of odorous air, the odor present in the mixture is reduced to threshold. The ratio of the dilution air, odorous air mixture to the odorous air is referred to as the number of thresholds of a given odor present.

This apparatus employs the same principle used in the Camera Inodorata by Zwaardemaker[18] and in the commercial Osmoscope by Fair and Wells[19]. Various other instruments of the same type[20-24] have found application in the perfumery, water supply and sewage disposal industries. Upon the advice of E. R. Weaver of the National Bureau of Standards, an instrument (see Fig. 1) was designed, modifying this principle for use in practical odor measurement.

In Fig. 1, the odorous air is drawn from test area through tubing A by blower B. This air quantity is measured by rotameter C. The odorous air passes to

mixing chamber D where it is mixed with the purified dilution air. Dilution air is supplied by passing compressed air through a purifying train E for removing impurities[17]. This air is then measured by rotameter F before entering mixing chamber D. The air mixture is discharged at sampling orifice G. The operator records the air flows at which the odor is just detectable at G. By proper balancing of air flows, this presents a method of reducing any concentrations of odor to terms of thresholds. For example, if 2 cfm of purified air mixed with 1 cfm odorous air renders the odor barely detectable, the odor in the original air was at a concentration of 3 thresholds.

Because of its dependence upon the acuity of the olfactory sense, this technique, uncontrolled, would be open to question from two angles. First, since the sensory sensitivity may vary within an individual from time to time, comparable odor measurements could not be taken by a given operator over

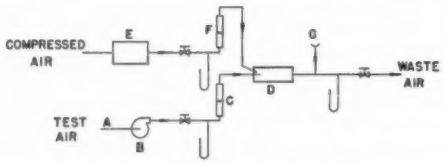


FIG. 1. DILUTION SYSTEM TO MEASURE THRESHOLDS

prolonged periods of time. Second, the odor levels taken by one worker do not necessarily compare with those of another.

In the first case, standardizing of the operator, two techniques have been devised. In surveying a large group of operators for this work, while extreme variation in individual sensitivity was found, two operators were found to have practically identical sensitivities to tobacco odors, cooking odors, and certain synthetic odors. Since it is unlikely that two individuals' sensory perception would vary simultaneously from physiological causes, two such operators served as a control for each other. During test runs, when the readings of the two coincide, the data were considered valid; when the readings were divergent, the data were discarded. It was found that in almost 100 percent of such readings over a four-year period, the two operators confirmed each other so that reports of variation of the human sense of smell appear to be exaggerated.

Where two controlling operators are unavailable, another technique was found suitable for establishing absolute baseline of sensitivity. This involves the dilution measurement of a constant known odor to determine the percent efficiency of the nose at any time during a series of experiments. In an air-tight container, xylene was permitted to permeate the air to saturation under controlled temperature and pressure conditions. In a prolonged series of experiments, the normal odor level of xylene was found to be 380 thresholds. If, on a given day, an operator found xylene threshold values greater or less than 380, his percent

efficiency could be calculated and his test data corrected accordingly. Divergent readings of two operators can be corrected in a similar manner.

Physical Measurement: Physical measurement of odors has been attempted by various light absorption instruments. None of these were sufficiently sensitive to indicate odor concentration.

Chemical Measurement: It was found that certain oxidants react with odor complexes in a reproducible manner. One in particular was completely interreactible with tobacco and cooking odors. During this reaction, the oxidant is reduced and the odors destroyed. The amount of reduction is measurable and directly proportional to the quantity of odor destroyed.

Three major systems were used in the chemical study of tobacco smoke characteristics. While in earlier studies, these systems were distinct and separate, their principles are discussed in terms of Fig. 2, which is a composite of all three.

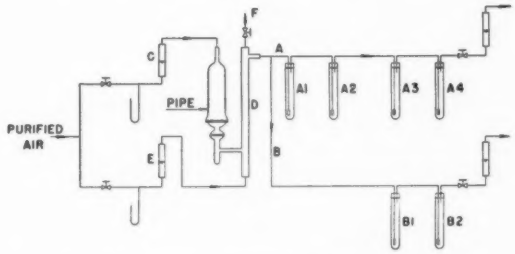


FIG. 2. ABSORPTION SYSTEM USED IN CHEMICAL MEASUREMENT OF ODORS

In the first studies, tobacco was burned in the pipe in the presence of purified air at various flow rates measured by rotameter C. All of the tobacco smoke passed through the mixing chamber D and absorption train A, train B being eliminated from the system by valves. (The absorption trains are a series of 1-5 tubes, each tube containing an oxidant with which the odorous air is brought in intimate contact by gas dispersion tips). The results were recorded in terms of quantity of oxidant reduced per weight of tobacco burned. When studying the effects of dilution on tobacco odor, quantities of purified air, measured by rotameter E, were passed into the mixing chamber D, together with odorous air from the pipe. Portions of this air passed through train A and the remainder was discharged to waste F.

For measuring deodorant abilities equal portions of diluted odorous air were distributed to each of the two trains, A and B. In train B the odorous air passed directly into two tubes, B-1 and B-2, containing the oxidant for measuring the quantity of odor present in the original air. In train A, the odorous air passed through the deodorant mechanism under test (A-1 and A-2) before passing to the oxidant measurement tubes (A-3 and A-4). This measured the quantity of odor left in the air stream after subjecting it to the deodorant. A comparison of the quantity of oxidant reduced in each of the two legs indicated the efficiency of the deodorant.

Experimental Results: With the development of means of measuring basic deodorant ability (oxidant absorption), and evaluating practical results (dilution threshold system) the way is clear for a standard evaluation of air deodorants. By these techniques and toward this goal certain essential fundamental knowledge has been gained.

Using the oxidant train, quantitative measurements of the odorous components of tobacco smoke were determined (Table 1). In these tests, four rates of combustion were used; the slowest approximating the smouldering rate and the most rapid approximating the puffing rate of natural smoking. In conditions 1 and 2 all of the odor produced by the complete burning of the cigarette was collected in the absorption train. In conditions 3 and 4 the odorous air was diluted 30 times by purified air and only 1/100 of the total mixture passed

TABLE 1—QUANTITY OF OXIDANT REDUCED BY THE VAPORS OF BURNING TOBACCO

CONDITION	BURNING TIME 1 CIGARETTE MIN	ML N/10 OXIDANT REDUCED PER GRAM TOBACCO BURNED
1. Total Absorption	15	120
2. Total Absorption	35	125
3. Dilution Partial Sampling	18	118
4. Dilution Partial Sampling	0.5-1	117

through the absorption train. In these tests various quantities of name cigarettes, both fresh and stale, were used.

In all cases the results were strikingly similar as recorded in Table 1 which is a composite of 200 tests. There is close uniformity in odor production by cigarette tobacco regardless of brand, degree of freshness, rate of combustion and this holds true even with extreme dilution of the odor. Artificial burning of tobacco, because it is reproducible, can therefore be used as a laboratory tool in the investigating of odors; and the variables mentioned which would be expected in human smoking need not be considered as interfering variables. In comparing mechanical with human smoking in normal office spaces, it was found that a certain constant correction factor must be used because of the absorption of the odor in the respiratory tract.

This technique permits the laboratory determination of the efficiency of a deodorant against mechanically produced tobacco smoke odor. By the proper chemical analyses and the correction factors the quantity of the deodorant consumed, hence the cost of a deodorant process, can be predicted in a practical installation on the basis of anticipated amount of smoking.

The simple knowledge of the total quantity of deodorant necessary to remove completely a given quantity of odor is not sufficient. It involves the balancing of production of odor against the removal of odor by the deodorant. This will vary with the contact of the odorous air with the deodorant, the rate of odor production, and the odor levels involved.

To explore this it was necessary to work with actual smokers in normal rooms. Using four smokers at a controlled rate of smoking without fresh air makeup

or a deodorant mechanism, the odor in the test room builds up in regular steps to intolerable levels. This odor gradually built up to a maximum of 14 thresholds. There is a distinct relationship between quantity of odor produced per cubic foot of air and threshold.

In this test it was found that a group of assorted subjects, smokers and non-smokers, recorded the room as immediately objectionable when concentrations of 2.5 to 3.5 thresholds were reached. In a room maintained at 2.0 thresholds, a similar group of subjects reported the condition distinctly bad after prolonged exposure. It has been established the tobacco odor in any room should be held at or below 1.3 thresholds.

Successful deodorization experiments demonstrating these principles have been run in recirculating type systems wherein odorous air is removed from a room, passed through a deodorizing chamber, and returned to the room. The principle is one of dilution as in standard air conditioning, substituting the deodorant for the outside air makeup. Just as in normal ventilation, when any odor source (burning tobacco) is present, it is impossible to prevent the odor from being detected no matter how rapid the recirculation. The objective is rather to prevent this odor from exceeding the objectionable level. To do this the odor must be removed at the same rate as it is produced. Since the only odor which can be removed is that which is carried to the deodorant by the recirculating air, the quantity of this air, the concentration of odor in it, and the ability of the deodorant to reduce this odor are all contributing factors to the efficiency of a deodorant installation.

This is illustrated by a series of experiments run in a test office space. The occupancy was four men, each smoking one-half pack cigarettes per 8-hour day, or 3.3 gm tobacco per hour (mechanical smoking was also used). Within less than 1 hour, the threshold would rise to objectionable levels of 1.9 thresholds; exceeding the desirable 1.3. Once the threshold level had reached 1.3, odor had to be removed from the air at the rate of 3.3 gms tobacco burned odor per hour. Since the only odor which could be removed was that carried to the deodorant at a concentration of 1.3 thresholds, the quantity of air to be recirculated depended upon the efficiency of the deodorant. It was impossible to find a 100 percent efficient deodorant, but substituting outside air for a deodorant (equivalent to a reduction of 1.3 to 0 per pass), the outside air required was 42 cfm. In pursuing these tests with deodorants less than 100 percent efficient, one which would reduce the odor level in recirculated air from 1.3 to 0.5 thresholds per pass through the deodorant would require the recirculation of 69 cfm and one reducing from 1.3 to 0.7 would require 90 cfm.

By knowing occupancy, smoking rate, and deodorant efficiency, the specifications for any size installation can be predicted.

BACTERIA

Experimental Procedure: The degree of importance of air-borne infection is still unestablished in the long term contention between *air-borne* and *contact infection*[25]. A realistic view must be taken in the establishing of logical requirements.

There is ample evidence as to the existence of air-borne infection[9, 11] as summarized in any bacteriological text. Since the possibility of air-borne infec-

tion is dependent upon the number and virulence of the infecting organism and the state of health of the host, it is impossible, with current knowledge, to set any numerical limit above which infection will occur and below which infection will not occur. Therefore, while standards of the nature used in water and milk remain for the future, an interim standard is recommended.

Current ventilating practice is based on the premise of using quantities of outside air as a means of keeping inside air sanitarily acceptable. Such a procedure presumes that, because of confinement, increase of bacteria within an enclosed space will increase the likelihood of infection. Also, because outside air is relatively free of infecting organisms, ventilation can, by dilution, reduce the likelihood of infection. Therefore, it has been accepted that outside air is desirable and inside air is less so, without any knowledge of degree. The goal to be attained is the rendering of inside bad air comparable to outside good air. Because, in general, the total bacteria counts of inside air are much higher than those of ideal outside air and because of the lack of knowledge as to the comparative numbers of pathogenic bacteria, inside and out, total bacteria counts become the yardstick.

For an air sanitizing agent to have merit, it must reduce inside total bacteria counts to concentrations approaching outside counts. Because there is no constant outside or inside count, this must be a flexible comparison. Therefore, each air sanitizing installation must be a comparison of three conditions; inside bacteria level before treatment and following treatment, both being controlled by outside counts.

The acceptance of such a procedure would demand the use of standard sampling devices. Such devices for air-borne bacteria are as confused as air sanitation itself[13]. In recent correspondence from the U. S. Public Health Service, the following comments were made:

> In very broad and general terms, I could only suggest that if you are sampling extremely large particles (20-200 microns) settling plates might be useful. For medium size particles (3-20 microns) a sieve sampler is probably more satisfactory, while for extremely small particles (0.3-3 microns) one of the liquid impingers would probably be best.

Because the proposed results will be comparative in nature the absolute efficiency of the measuring devices becomes relatively unimportant.

Simplicity and ease of operation prompted the use of a centrifugal impingement device[26] for this comparative investigation of bacteria levels though high efficiency has been demonstrated by some[27-29] and discredited by others[30, 31]. Five samples taken over a period of one hour at a rate of 1 cfm for 10 min yield statistically valid average counts. Samples were collected on nutrient agar (5 percent blood) and incubated at 98.6 F for 48 hours in an effort to eliminate many of the saprophytes and to encourage the growth of those capable of inhabiting an animal host. Records were kept of number of total organisms per unit volume of air and also broken down into hemolytic organisms and molds to develop these types as a potential sanitary yardstick.

In the foregoing proposition there is one fallacy. In using simple reduction of inside air counts as an indicator of sanitary merit it is possible that the reduction of total counts is not reflected by a corresponding reduction in pathogenic counts. The vast majority of air bacteria are perfectly harmless.

Therefore, over and above the reduction of room counts it must be demonstrated that a disinfectant mechanism has a specific effect against air-borne pathogens. Three pathogens which appear most frequently in the literature as common air infectors were chosen. These organisms secured from the American Type Culture Collection are *Corynebacterium diphtheriae 296 Staphylococcus aureus* 6538, and *Streptococcus pyogenes* 4543 *(Micrococcus pyogenes var. aureus)*.

In a practical installation such as a washer, it was visualized that an organism could be picked up by the wash water, and, in one recirculation, be returned to the air stream quickly enough to be unaffected by a disinfectant. Therefore, a disinfectant must be of such strength as to destroy these pathogens within one recirculation time of the wash water. In most applications, this time was found to be 40 sec. Therefore, in comparing disinfectants, and in setting minimum concentrations of disinfectants, 95 percent kill of these three organisms is

TABLE 2—MICRO-ORGANISMS PER 10 CU FT OF AIR AT DIFFERENT LOCATIONS AND CONDITIONS

LOCATION	TOTAL ORGANISMS	RATIO TO OUTSIDE AIR	MOLD	HEMOLYTES
Outside	12	—	1	1
Inside (Not Ventilated)	99	8.3	3	7
Inside (Ventilated)	55	4.6	3	4
Washer (Water—clean)	66	5.5	7	4
Washer (Water—dirty)	185	16.0	6.3	6.3
Washer (Chemically treated)	20	1.6	0.7	1.3

required before plant testing. If the proposed application is water, or another liquid, washed inocula are placed in tubes of these solutions under properly controlled temperature conditions for 40 sec. Percent kill is determined by standard dilution plate counts on the proper media, with proper control.

Experimental Results: Bacteria work has been directed toward the establishment of a consistent difference between inside and outside air. Over a 5-year period more than 5000 air samples have resulted in the average counts recorded in Table 2. This study has included counts primarily in buildings of the office and hotel types in and near business districts.

As indicated in Table 2, inside counts are 5 to 16 times higher than outside counts. Spaces which are not ventilated (*i.e.*, no outside air) are 8 times greater than outside, while in those which are ventilated (dry coil systems) the outside air has diluted the inside bacterial loads to 4.6 times outside. Counts in spaces conditioned by washers freshly cleaned, approach the dry coil loads with a ratio of 5.5 while in dirty washers the ratio is 16, even with normal outside air.

The results of a chemically treated washer are also included. A disinfectant chemical under test was added to a dirty washer without outside air. The room air dropped to, and was maintained at, 1.6 times the outside level. Almost reaching the ideal outside count this chemical demonstrates its ability to substitute for outside air. Since it was first demonstrated that this chemical was disinfectant against the air-borne pathogens, it would be expected that this 90 percent reduction in total micro-flora would be reflected in similar reduction

of pathogens. Similar extreme reduction of hemolytes and molds are shown, and indicate a non-preferential reduction of air-borne bacteria.

OTHER CONSIDERATIONS

Many disinfectant and deodorant chemicals like those discussed have been investigated. Referring only to the disinfectant-deodorant techniques described, many have passed the requirements. However, there are other considerations which must be taken into account before a sanitizing mechanism can become a commercial product. The following are the requirements which have been found in an effort to further characterize air sanitizing agents:

1. It must be odorless and create no offensive odor.
2. It must be non-toxic.
3. It must be non-flammable and create no fire hazard.
4. It must be non-corrosive.
5. The mechanism must be understood as insurance against unknowing failure of the first four requirements.
6. The field control of this mechanism must be such as to be operated by non-technical personnel.
7. The mechanism must be such as to be free of undesirable end products.

CONCLUSIONS

It has been pointed out that air conditioning odor and bacteria control literature, though voluminous, has been of such individual and uncorrelatable nature, that at this time not only is there no program by which this subject can be pursued, but also there are no accepted basic methods by which the results of such a research program can be uniformly evaluated by this industry.

Basic techniques are described herein by which the air sanitation problem can be advanced more quickly by the securing of experimental results in fundamental terms. By the removal of the human descriptive element, the practical evaluation of progress and the comparison of results throughout the field are possible.

It has been shown that such studies must involve naturally occurring odor complexes, but that the control of such odors as burning tobacco, and cooking odors, is possible. Concentrations of tobacco odor are uniform and measurable, and, therefore, the deodorant effect of any chemical or mechanism can be quickly evaluated from the standpoint of efficiency, cost, and other limiting factors.

By the dilution-threshold method, odor results in a practical plant can be evaluated. By the establishment of odor loads, desirable odor levels, and odor removal efficiencies, the requirements of a deodorant are clarified and means for satisfying these requirements are given. It permits the solution of odor problems in an engineering manner and presents bases from which to specify for any installation.

Although there is no standard of bacterial air sanitation, it is demonstrated that there is a consistent difference between inside and outside air, and, by comparative measurement of inside and outside air counts, the degree of effectiveness of any mechanism for reducing bacteria levels can be determined. To

be assured that a reduction in total room counts reflects a reduction of pathogen count, simplified disinfectant tests on certain air-borne infectors are recommended.

Satisfactory air sanitizing agents must not only be satisfactory from a disinfectant-deodorant viewpoint but must also be inoffensive, safe, non-corrosive and subject to control as already pointed out.

ACKNOWLEDGMENTS

Appreciation is expressed to J. D. Karnes and F. C. Miller of the Research Department of York Corp., for their valuable assistance in this work.

BIBLIOGRAPHY

1. Air Sanitation, by J. M. Dalla Valle (*Heating & Ventilating*, May 1949, pp. 57-71).

2. *Preventive Medicine and Hygiene*, by M. J. Rosenau (Sixth Edition, D. Appleton-Century, 1935).

3. Organic Matter in the Expired Breath, by M. J. Rosenau and H. L. Amoss (*Journal Medical Research*, 1911, p. 35).

4. Zehn Jahre Forshung Auf Dem Physikalisch-Medizinischen Grenzgebiet, by F. Dessauer (Leipzig, Georg Thieme, 1931).

5. Observations on Group of Subjects Before, During, and After Exposure to Ionized Air, by C. P. Yaglou, et al (*Journal Industrial Hygiene*, September 1933, pp. 341-53).

6. Environment and Health, U. S. Public Health Service, Federal Security Agency 1951.

7. Minimal Replenishment Air Required for Living Spaces, by W. V. Consolazio and L. J. Pecora (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 127).

8. Odors and the Sense of Smell (A Bibliography by Airkem, New York, 1952).

9. Aerobiology, by F. K. Moulton (*American Association for the Advancement of Science*, Washington, 1942).

10. *Odors, Physiology and Control*, by C. P. McCord (First Edition, McGraw-Hill Book Co., Inc., New York, 1949).

11. Studies in Air Hygiene (*Medical Research Council, Special Report, Series No. 262*, London, 1948).

12. *Experimental Air-Borne Infection*, by T. Rosebury (Williams & Wilkins Co., Baltimore, 1947).

13. *Flavor*, by E. C. Crocker (First Edition, McGraw-Hill Book Co., Inc., New York, 1945).

14. *The Chemical Senses*, by R. W. Moncrieff (Second Edition, Leonard Hill Ltd., London, 1951).

15. Henning's Olfactive Prism and the Preparation of Perfume Bases, by O. Gerhardt (*Parfumerie Moderne*, 1928, pp. 415-23).

16. Seeking a Working Language for Odors and Flavors, by E. C. Crocker (*Industrial and Engineering Chemistry*, 1935, pp. 1225-28).

17. The Effects of Moisture Content on the Diffusion of Odors in the Air, by R. L. Kuehner (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 77).

18. A Camera Inodorata, by H. Zwaardemaker (*Perfumery Essential Oil Record*, 1921, 12:243-44).

19. Determining Odors in Water, by G. M. Fair and W. F. Wells (*Water Works Engineering*, 1934, 87:1051-55).

20. Apparatus for Determining Odor in Water, by W. B. Hart (*Industrial and Engineering Chemistry*, May 1937, pp. 243-46).

21. ABC of Taste and Odor Control with Activated Carbon, by E. A. Sigworth (*Public Works*, September 1940, pp. 33-34, 37-40).

22. Studies on Accuracy of Threshold Odor Values, by R. Hulbert and D. Feben (*Journal American Water Works*, November 1941, pp. 1945-64).

23. Odor Identification of Chemical Warfare Agents, by E. M. Hoshall (*Chemical Warfare Bulletin*, July 1941, pp. 99-107).

24. Calculation of Threshold Odor, by H. A. Thomas (*Journal American Water Works Association*, June 1943, pp. 751-69).

25. *Rosenau Preventive Medicine and Hygiene*, by K. F. Maxey (Seventh Edition, Appleton-Century-Crofts, New York, 1951).

26. Apparatus for the Study of the Bacterial Behavior of Air, by W. F. Wells (*American Journal Public Health*, 1933, pp. 58-59).

27. The Effectiveness of Certain Types of Commercial Air Filters Against Bacteria (B. Subtilis), by J. M. Dalla Valle and A. Hollaender (*U. S. Public Health Report*, 1939, pp. 695-99).

28. A Simple Device for Sampling Air-Borne Bacteria, by A. Hollaender and J. M. Dalla Valle (*U. S. Public Health Report*, 1939, pp. 574-77).

29. Electrostatic Method for Collecting Bacteria from Air, by C. M. Berry (*U. S. Public Health Report*, 1941, pp. 2044-51).

30. Hospital Masks: Their Bacterial Efficiency and Resistance to Air Flow—A Comparative Study, by R. Rooks, L. J. Cralley, and M. E. Barnes (*U. S. Public Health Report*, 1941, pp. 1411-19).

31. A Study of the Performance of the Wells Air Centrifuge, by E. B. Phelps and Leo Buchbinder (*Journal of Bacteriology*, 1941, pp. 321-44).

DISCUSSION

H. C. MURPHY, Carmel, Calif. (WRITTEN): Dr. Kuehner's paper is a definite contribution to our work in establishing standards in two important phases of air sanitation. However, in addition to the study of bacterial and odor contamination, I believe we will eventually find it desirable to further investigate the electrical properties of the lower atmosphere.

Dr. Theodore Puck of the University of Colorado Medical Center reported to the 1950 meeting of the *American Chemical Society* that viruses apparently must carry

an electrical charge in order to attack living cells. It appears that if the cells already carry a certain charge, the virus is incapable of becoming attached to it.

We know too little about the electrical characteristics of the air we breathe. In most of the studies reported, it appears that positive ions outnumber the negative ions, especially in places of public assemblage. It is possible that electric heaters, electrostatic air cleaners and similar devices which liberate positive and negative ions, may significantly alter the ionic content of occupied spaces. A number of investigators in this country and abroad are of the opinion that a preponderance of positive ions in the air we breathe has a depressing effect on the occupants while negative ions give a feeling of well being. Work on this subject is under way at several schools in this country.

R. G. H. SIU, Washington, D. C. (WRITTEN): The paper by Dr. Kuehner has analyzed quite clearly the difficulties connected with measuring the effectiveness of bacterial and odor control. He has done a very good job in pointing out areas to which energies should be directed.

The general procedure outlined for quantitatively expressing the amount of odor by *thresholds* is sound. Yet I believe it should be related to a definite quantitative value for a given odor. To say that *the odor concentration in the original air was 2 thresholds as compared to one threshold in the purified air,* provides a relative measure of the effectiveness of the purification processes. However, it would be difficult to compare the effectiveness measured by different individuals *unless* the individuals respond to the same thresholds.

Let us take an illustrative example: If test subject A is sensitive to 4 ppm of H_2S, the amount of H_2S removed in the example given will equal 8 minus 4, or 4 ppm. If test subject B is sensitive to 1 ppm of H_2S, then the amount of H_2S removed in the same example would be 2 minus one, or 1 ppm. In both cases the concentrations were reduced from 2 to 1 threshold. But the quantity of H_2S removed (and hence the purification efficiency) varied by a factor of four. It seems as if the test subjects themselves must first be calibrated against absolute amounts of odor material so that only those individuals sensitive to a standard threshold concentration will be used.

An interesting point was brought up by Dr. Kuehner in using xylene to check the sensitivity of its subjects. In the selection of the chemicals for such calibration, cognizance must be paid to avoiding those which numb the smelling senses. Some of the hydrocarbons have that capacity. I am not sure whether xylene belongs to the group.

The section on bacterial control has also been well done. The difficulties in adopting a standard of measurement in bacterial control resemble those in odor in that in both cases there are many different varieties. Dr. Kuehner pointed out, and rightly so, that attention must be given not only to the measurement of bacterial population as such, but specifically the pathogenic type and he mentions the diphtheria and pus-forming organisms. I was wondering whether the engineering profession, at least at the start of these investigations, should not be content to standardize on methods in which total bacterial counts are made, rather than venture into the difficult task of separation of organisms and identification of specific types. Furthermore, the selection of the particular pathogen to be considered would require quite a bit of deliberation.

If total bacterial population is to be determined, the use of simplified manometric procedures might be considered worthwhile as a substitute for plate counts on agar. The manometric technique using respiration as an index of growth has been worked over quite a bit in other fields, and consequently the time is ripe for the engineering profession to exploit the knowledge.

I believe the most immediate action called for by Dr. Kuehner's paper is a tentative agreement by the profession on at least one method as a point of departure. Dr.

Kuehner's paper shows very well that such a method can be described at this time, provided the desire is there.

N. E. BERRY, Evansville, Ind. (WRITTEN): In the preparation and presentation of this paper Dr. Kuehner has made a significant contribution to the long-term progress of the air conditioning industry. His description and discussion of the basic techniques available for bacteria and odor control work should be of positive value to those already active in this field and should provide a stimulus to others to initiate research work of this type.

I want to particularly congratulate the author on his objectivity in evaluating the accuracy of the techniques and of the data obtained. It is only through such painstaking research that we shall ever get solid ground under our feet in this phase of the industry. It is to be hoped that investigation of other possible components of air quality will be conducted on a similarly high scientific level.

With regard to the data presented, it occurred to me that there might be some misunderstanding of the results presented in Table 1. The third column is labeled, ml N/10 Oxidant Reduced per gram Tobacco Burned. In the case of total absorption the results are quite straightforward. However, in the two latter cases where partial sampling was used, the values are apparently calculated, taking into account the known percentage of the total smoke odor that was actually passed through the absorption train. Perhaps this is too obvious to deserve comment, although I must confess it was not immediately apparent to me on first reading.

Dr. Kuehner does not tell us how to solve our odor problems, but he does give us some valuable insight into what is going to be necessary in the way of research if we are to solve them. I have no doubt that this paper will be of particular interest to those in our Society's Research Laboratory who are just starting this type of investigation.

Although my personal inclinations are often toward basic research work with very precise results as the ultimate objective, I cannot resist this opportunity to emphasize the need for at least partial practical answers as we go along. Perhaps it is too much to expect both kinds of result from the same investigative group, although I believe something of this sort should and will be achieved in the A.S.H.V.E. laboratory work.

In conclusion I would like to repeat by way of emphasis, one sentence in Dr. Kuehner's paper. *The future demands that this industry assume the responsibility for air conditioning in its broadest sense.* I believe with the author that we must accept this responsibility, not necessarily as a duty to our fellow man, but as a challenging business opportunity. One does not have to be prophetic to know that in the years to come we are going to be doing things to air that we've not yet dreamed of. Certainly then we must move forward with all possible speed to practical solutions for those problems we have with us today and which we can all readily recognize without resort to dreaming.

G. W. MEEK, New York, N. Y. (WRITTEN): Both the author and his employer are to be congratulated—the author for the scholarly and painstaking manner in which he has carried out his studies, and his employer for having the vision and courage to finance work in a phase of air conditioning where ignorance is so extensive that the more timid are reluctant to enter.

It is unfortunate, however, that the paper is put forth in the form of a *teaser*—that the important findings of this research work are only suggested. As the author's bibliography shows, dilution systems and threshold measurement techniques are well known. It is also well known that certain foul smelling chemicals and certain highly corrosive chemicals have been used to kill airborne bacteria. Certain chemicals capable of partially oxidizing odorous molecules are also known. But the paper hints that

non-odorous, non-corrosive bacteriacides and non-odorous partially effective odor oxidants have been found. Without information on these materials, which supposedly were isolated and their performance tested, this paper provides its readers with no means of checking its scientific validity. Its main value seems to be that of an interest stimulator. It merely presents to the reader a challenge to attempt to learn more in nine years than the author's investigations disclosed, but the author will not tell all he discovered.

Three other points require comment. First, in his opening the author says, *This discussion applies to the absolute removal and destruction of bacteria and odor.* It is well to remind ourselves that what we so fondly call *fresh* air is *not* odor-free air. It is air which contains odorous molecules providing that subtle psychological effect which the human brain has learned to associate with the areas where such odorous molecules are generated—namely the great outdoors. If we remove these desirable odorous molecules and don't restore them we duplicate an error made recently by our medical colleagues. They suddenly discovered that Terramycin was so effective that it killed off all the *favorable* flora and fauna in the large intestine and that some tens of millions of friendly bacteria had to be replanted by having the patient drink a copious supply of buttermilk.

Second, certain of the author's opening remarks about odor removers and odor neutralizers might be taken to mean he feels there is no such phenomena as odor or osmic neutralization. I am sure this is not the case, because the very scientist on whose ideas the author's dilution apparatus is based showed osmic neutralization to be a scientific fact and completely devoid of any implications of masking or anesthetizing.

Third, the author's emphasis on absolute destruction of odor might imply that the day of odor-free air conditioned spaces has been made practical by this new discovery. We need but recall how odors are generated to know that such an accomplishment is not even theoretically achievable with solely a subtractive or purely a dilution approach to odor control. Recall that in an airplane or pullman car (two of our more smelly air-conditioned spaces) odor is *constantly* generated in the immediate vicinity of the occupant—tobacco smoke, odors from the surfaces and the openings of the human body itself, vomit, cosmetics, etc. Any system which requires that all of these odorous molecules be transported back to some central conditioner must of necessity continue to drag hundreds of millions of these same obnoxious molecules past the noses of the occupants.

H. B. NOTTAGE, Los Angeles, Calif. (WRITTEN): Have other arrangements of dilution systems been tried for measuring thresholds? If so, are the measurements independent of the apparatus?

Were any odors encountered which gave anomalous results by the dilution technique when compared with fullscale room measurements? If not, might this occur? Could the dilution technique be employed, for instance, to study the effect of humidity upon odors?

AMOS TURK, Danbury, Conn. (WRITTEN): A report of research results, if it is to fulfill its function as a contribution to knowledge, must be *operational* in nature. That is, other workers, *skilled in the art,* must be able to reproduce the research in their own laboratory, to confirm its value and to use it as a spring-board for further progress. This criterion of reportable research, development, or invention is used by the United States Patent Office and by editors of scientific journals in general. It is regrettable that this standard was not applied to this paper.

For example, the author describes a system employing an *oxidant for measuring the quantity of odor present in the . . . air.* The *concept* of oxidizing odorous vapors for analysis is old and does not merit republication, except in a review article. The use of new oxidants or new procedures, however, is certainly valuable and should

be published. The author probably worked with some novel procedures, and if so, their *operational* description should have been given. Also, what *deodorants* were evaluated? All of the experimental procedures and results reported in the paper are operationally deficient.

A number of incorrect statements appear in the sections on odor . . . *there is no satisfactory progress toward their* [*odors*] *control* and *the air conditioning industry could not now by any single criterion, or group of criteria, determine which deodorant or disinfectant is better than the other, or if they have any merit at all.* On the contrary, applications of activated carbon physical and chemi-sorption, catalytic combustion, and liquid absorption, with and without chemical reaction, are all well known and certainly constitute progress toward their control. Analytic methods for evaluating efficiencies of these techniques are abundantly reported in the chemical analytical literature, and are available to the air conditioning industry. Odor dilution methods, instruments, and procedures are also described, including a recently developed modification (*Air Repair*, Vol. 1, No. 4, 13, 1952). The literature on evaluation of disinfectants is likewise abundant.

The author confuses *odor* with *odorant*, the one being a psychological stimulus, the other a substance capable of producing such a stimulus. In this light reexamine the concept of an *oxidant for measuring the quantity of odor present in the . . . air.* All that an oxidant can *measure* is a reductant. The assumption that the chemically reducing properties of substances are directly related to their odorous qualities is incorrect, often astonishingly so. From literature reported to date, the problem of purely instrumental measurement of *odor* is still unsolved. It may well remain unsolved until the mechanism of odor perception is better understood.

SHERMAN ROSS, College Park, Md. (WRITTEN): The techniques the author described are of interest, and the development of these should be of importance in the future. I doubt that we have here the *yard stick* required.

I am quite sure that Dr. Kuehner was concerned with techniques which could be conducted *in an engineering manner* to deal with this complex problem of human sensitivity. I am not at all sure that he has avoided the major and determining variable of human sensitivity and human preference. He is searching for measurement in finite numbers with which the human factor is minimized and quantified. It is precisely this human factor for which all of the odor removal is done. In attempting to remove two important sources of variation, Dr. Kuehner makes several major assumptions and possibly several major errors. He is concerned with the removal of individual differences, and of dealing with results from observers whose agreement is extremely high.

In the dilution system he presents for measuring thresholds, we are not informed of the problem of sensory adaptation, nor do we know to what degree his thresholds were influenced by the stream of diluted material used. He does not inform us of the range of individual differences which he encounters in measuring thresholds by his system or the basis for the selection of two operators (except their thresholds agreed when measured for several odor systems). We do not know whether this sample ($n = 2$) represents high, middle, or low sensitivity when compared with the population. We might note in passing that two completely anosmic individuals would agree perfectly in the sensitivity to the various substances used. In spite of this sampling situation, Dr. Kuehner does not hesitate to suggest that *variations of the human sense of smell appear to be exaggerated.*

Regarding the use of measurements from the two *standard observers*, we might comment that the treatment of these results as described is certainly unusual. The implications of discarding *divergent readings* from these observers could only bias the results used in the direction of increased reliability.

Dr. Kuehner does not indicate to us how test data are corrected when an operator's current threshold is found to be different from his usual threshold. It might be that he assumes a linear relationship between percent of correct judgments and concentration. Most certainly the notion of a linear scale here is wide open to question. It is also an interesting question to ponder on the meaning of a statement that an observer was 110 percent efficient this morning.

Dr. Kuehner has tried to get rid of complex variables, namely, individual differences in sensitivity and variability in sensitivity. It is precisely these variables in the whole measurement situation which are the most significant, and it is toward human behavior that the whole air-conditioning industry directs its efforts.

The techniques, problems, and difficulties of experimental psychological study simply cannot be assumed out of the situation, nor mistreated conceptually, when we have such a major backlog of information of useful importance.

I have tried out these comments on several of my colleagues who agree with their general flavor. I feel confident that other psychologists would concur in the foregoing opinions.

L. T. AVERY, Cleveland, Ohio: Mr. Chairman, I cannot miss this opportunity to remind you that the Society has been criticized at times for venturing into fields other than straight engineering application.

As I visited the chapters and talked with topflight scientists and, where I could, physiologists and bacteriologists, it was very definitely questioned whether we had any right to be concerned with the physiological, biological or other aspects of air conditioning beyond the simple one of temperature and humidity control. Air cleanliness was taken for granted as being a necessary evil.

I suggest to you, as I have suggested in the past, that you cannot avoid these co-relationships. Your Society has, therefore, proceeded with your money in its research work and has built two rooms, which Mr. Meek mentioned, at the A.S.H.V.E. Research Laboratory. No tests have yet been run, but work is now getting underway.

We are studying the physiological relationship of man to his environment for shock effect, the most recent report being given in the paper presented by the University of Illinois at this meeting*. I think it very appropriate to recognize as engineers that we cannot avoid responsibility in this relationship.

I suggest to you also that if we do not accept responsibility in this matter it will be brought to us and laid on our doorstep, as have other aspects of our business, through legislation. Certainly, if any legislation is to be prepared with regard to odors and sanitation, we should have a definite part in preparing the standards or in establishing the minimum permissible requirements.

The Society, in cooperation with several other organizations now has a committee, with Walter Fleisher as chairman, rewriting the minimum requirements for comfort air conditioning. This is a tremendous task. Are we going to accept standards of perfection or standards of minimum tolerance? You cannot avoid odors and bacteria. Shall we remove both completely? It has been said that if you take out bacteria you make man more vulnerable because you have killed his friends as well as his enemies.

AUTHOR'S CLOSURE: What are the air qualities which affect man; in what way and to what degree are they desirable or undesirable; what are the measurement tools by which we evaluate them; how do we produce the desirable qualities? Concerning ionic characteristics of air, to which Mr. Murphy has referred, all four cited questions are as yet unanswered to our satisfaction. To establish some meaning we have been accumulating data during the past several years on the positive and negative ions in air, both inside and outside, occupied and unoccupied, conditioned and non-conditioned.

---

* A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment, by Tohru Inouye, F. K. Hick, R. W. Keeton, J. Losch and N. Glickman (p. 35).

So far we have been unable to establish a trend as to the desirable concentrations of either positive or negative ions or that they produce any practical effect at all.

Had this paper been a discussion of specific deodorants and their merits, the assigning of quantitative values to the thresholds would be a must. As Dr. Siu suggests, the simple statement of a threshold value would be meaningless to anyone except the specific operator. This is the theme of the paper and one of the two questions which has handicapped the successful use of the threshold system in the past. To be meaningful threshold values must be related to known and reproducible baselines. This is precisely why the paper stresses the constancy of odor production per quantity of tobacco burned. Results are considered in terms of the source producing the odor. The discussion on xylene (which does not anaesthetize) shows the second method for assigning relative quantitative values to thresholds by defining an operator's sensitivity.

The use of specific pathogens as an index of air pollution has been considered without success. Therefore reduction of total bacterial population is presented as the other alternative. In accepting this, however, one must recognize the specificity of disinfectant mechanisms. It is very possible that a disinfectant reducing the viable count in air might be selective to such a degree that the reduction is in saprophytes only. It must be demonstrated that a disinfectant under study is effective also against airborne pathogens. The three cited organisms were selected as typical pathogenic air contaminants. They are not suggested as indices of pollution.

Mr. Berry's comments are noted and are appreciated. There is oft-times confusion between *practical* and *basic* research. In investigations such as these involving a human sense, the distinction is even more difficult. While the work reported is of a fundamental nature, its value rests in the practical tools resulting therefrom. By these tools it is possible to evaluate deodorants even though we are still in ignorance of basic odor perception processes. Mr. Berry is correct in that the cited third column is calculated.

Mr. Meek has more than a suspicion that this is a *teaser* paper; that possibly we have discovered the perfect deodorant. Accordingly he seems to have girded himself with arguments against this hypothetical deodorant. Answers on the originality of my work will be discussed in my reply to Dr. Turk. The phrase, *absolute destruction of odor and bacteria,* only stresses that we search for a destructive device, not a masker in any of its forms. Odor maskers, including osmic neutralizers, certainly do exist as Mr. Meek says. With the psychological and control problems involved, we can't see their merit. Further, the addition of another air contaminant, however pleasant, defeats our goal of air sanitation. The achievement of a completely odor free and sterile air seems very unlikely. In fact, the dilution method as described demands a certain bacteria and odor level in the conditioned space. To us, the best approach is one of counter-balancing rate of production by an artificial removal. Accordingly, objectionable odors and bacteria will merely be reduced to acceptably pleasant levels.

Dr. Nottage has asked a number of questions. The design of the dilution system is relatively unimportant. We have used various blowers, air metering devices and mixing chambers. Provided the construction is not of a material very absorptive toward odor (*e.g.,* rubber) the measurements are independent of the apparatus.

Where a natural odor is a mixture of many component odorous materials, the cited anomaly may occur. We have not encountered it with tobacco or cooking odors. The dilution system could and is being used in the study of humidity effect on odors and odor perception.

In answer to Dr. Turk, I would mention that the method of presenting information and the extent of explanation in a technical paper is shaped to a degree by the audience. As long as we have scientific liberty any specialist acutely interested in the details of precise techniques can get them; their description to this Society in general would seem a waste of journal space to the average engineer.

In answer to both the Meek and Turk criticism of originality, a review of the paper will show that we make no claims for the novelty of the techniques and, indeed, give full credit to previous workers in the same line. By the same token, it would be redundant to describe more fully our particular apparati because they are modifications of well known techniques. By the special use of these techniques we feel we are illuminating Dr. Turk's concluding statements; that of putting odor concentrations in terms which are meaningful. In the absence of general agreement on such terms as *odor* and *odorant;* we have used the term *odor* as more generally recognized. Of course, an oxidant measures only a reductant. We have presented data which shows that a given quantity of tobacco burned yields a fixed physiological response (thresholds). The same quantity of tobacco burned yields a fixed reduction of oxidant. Axiomatically, then, physiological response can be put in terms of oxidant reduced. A loose use of nomenclature, perhaps, but valid tools.

By the use of these tools a discussion of odors and evaluation of deodorants can be put in other than the usual subjective terms. It was the inability to do this which has permitted chlorophyl to remain in an uncertain state for many years. Even odor specialists disagree as to the deodorant properties of this product. As long as this can happen, we apparently do not have satisfactory criteria for a deodorant and have not made satisfactory progress toward odor control.

It should be noted with reference to the comments made by Mr. Ross that we do not claim to have developed a psychological tool for studying olfactory perception. We do not suggest the calibration of all noses nor the development of universal standards of threshold values. Our instruments may have some use in psychology, I don't know. We developed them to evaluate, first in the laboratory and later in practical installations, the efficiency of many deodorants. These methods permit the comparison of results in numerical terms rather than "pleasant" or "less than pleasant, but not unpleasant". Secondly, by permitting preliminary laboratory screening, these methods eliminate the prohibitively time consuming statistical analyses of human reactions previously necessary in practical plant studies.

It goes without saying that in using the human nose, reproducibility is the major hurdle. Because we cannot control the sensitivity of the nose, we can do no more than develop methods of determining when the nose is at its normal sensitivity or above or below this point. In comparison of deodorants, one threshold must mean the same next week or next year as it does today. Earlier in our work, when an operator's sensitivity varied from his norm on a given day, he did not operate on that day, or his data were discarded. From a psychological standpoint the readings of an operator of uncertain sensitivity may be of some interest; they are not in our work. When we discovered later that we could correct mathematically when an operator was within 10 or 20 percent of normal sensitivity, the number of days of discarded data decreased.

For the purpose of this paper, the sensitivity of the operators described is unimportant. Dr. Ross has emphasized the point that this was not explained. Data were presented only to demonstrate that the nose can be calibrated to yield reproducible and useful information. To calibrate the nose, threshold must be related to some known and constant base. The obvious base is the odor source, *i.e.*, number of thresholds per gram of tobacco burned. For example, our operator's threshold of tobacco odor is 1.0 mg of tobacco burned per cubic foot of air. The relationship of chemical measurement is explained in the reply to Dr. Turk in the foregoing.

The statements of Mr. Avery typify what we believe to be the opinions of the majority of thinking people in air conditioning. There is no limit to the responsibility of an industry in its service to its customers. As to the standards to be attained, certainly perfection is to be desired, but we all realize that perfection must be modified by practicality. No method in the past has permitted the measurement of what is perfection or what compromises from perfection are economically necessary. While

the techniques described in the paper do not give absolute increments of olfactory sensitivity, nor methods of determining whether or not a given cubic foot of air is safe to breathe, they do give yardsticks whereby the degree, bacteria and odorwise, to which an air sanitizing agent improves or harms inside air conditions. Further, these techniques yield results which are understandable, meaningful and reproducible by the industry. They are presented with the conviction that great strides can be taken in this field and much service can be rendered to the customer even though the fundamental ignorances are as yet unanswered.

The question of removing the helpful bacteria along with the harmful has been discussed several times. The pros and cons of this are well rehearsed by O'Hara (*Airborne Infection,* Oxford University Press, 1943) and Stuart Mudd (*British Medical Journal,* II, 444, 1944). Like Bourdillon we ask, how many would eliminate chlorination of drinking water on this argument?

1467

No. 1467

HEAT TRANSMITTED TO THE I=B=R RESEARCH HOME FROM THE INSIDE CHIMNEY

By W. S. HARRIS* AND R. J. MARTIN**, URBANA, ILL.

WHEN DESIGNING a heating system for a house it is common practice to assume that room heat losses are a function of the indoor-outdoor temperature difference only, and that the sole source of heat input to each room is the room heating unit. While heat losses are primarily dependent upon the indoor-outdoor temperature difference, such factors as the direction and intensity of the wind, and the amount of sunshine also have an effect on the actual heat loss. Moreover, the room heating unit is not the only source of heat input to the rooms. In addition to heat supplied to the rooms by the room heating unit, heat is also obtained from such sources as lights, people, cooking, heating pipes and heat losses from the boiler and chimney. Early tests in the *I=B=R* Research Home showed that about as much heat was supplied to the rooms by these *stray* sources as by the room heating units. Furthermore, it was observed that those rooms through which the inside chimney passed were overheated even though the heat losses were estimated with extreme care and the radiation was selected in accordance with the estimated heat loss. To correct this unbalance in room temperature it was necessary to remove as much as 35 percent of the radiation originally installed in those rooms through which the chimney passed.

These observations seemed to indicate that a rather large amount of heat was being transmitted through the walls of the chimney to the room air. This paper is confined to a discussion of tests made in the *I=B=R* Research Home at the University of Illinois to determine the amount of heat supplied to the house in this way when the house was being heated by an oil-fired, forced-circulation hot water heating system. The results are compared with results of investigations made at the National Bureau of Standards and *Battelle Memorial Institute.* A method of allowing for the effect of chimney heat in the design of residential heating systems is suggested. Limitations of this paper do not permit a complete discussion of test procedure and computation methods; however, a more detailed

* Research professor of mechanical engineering, University of Illinois. Member of A.S.H.V.E.

** Associate professor of mechanical engineering and associate director of Engineering Experiment Station, University of Illinois.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

report has been published as University of Illinois Engineering Experiment Station Bulletin No. 407.

TEST EQUIPMENT

Construction details and the floor plans of the Research Home are shown in Fig. 1. The calculated coefficient of heat transmission for the wall section was

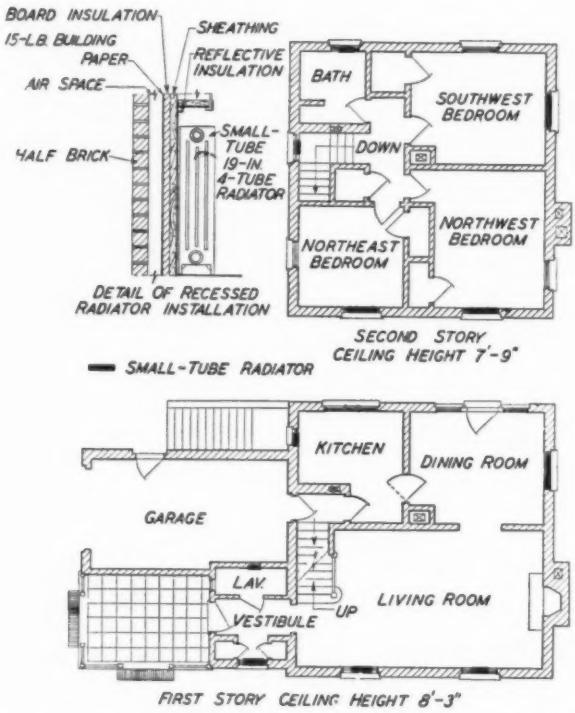


FIG. 1. CONSTRUCTION DETAILS AND FLOOR PLAN OF RESEARCH HOME

0.074 Btu per (hr) (sq ft) (F deg). All windows and doors were weather-stripped. The total calculated heat loss under design conditions of —10 F outdoors and 70 F indoors was 43,370 Btu per hr for the house, excluding the basement.

A one-pipe, forced-circulation, hot-water heating system was used. The oil-fired, cast-iron boiler had a net $I=B=R$ rating of 63,000 Btu per hr or a gross $I=B=R$ output of 95,000 Btu per hr when fired at a rate of 1.0 gal of oil per hr. Small-tube type 19-in., 4-tube, cast-iron radiators set in open recesses below windows, transferred the heat from the boiler water to the room air. The

radiators were connected to two supply loops located in the basement as shown in Fig. 2. Locations of all radiators are shown by the black rectangles in Fig. 1. The amount of radiation installed in each room was based on a heat emission rate of 200 Btu per sq ft EDR (Equivalent Direct Radiation) with a mean water temperature of 195 F, and was just sufficient to offset the calculated heat loss at an outdoor temperature of −10 F at these design conditions. The chimney to which the boiler was connected had a height of 35 ft 2 in., measured from the basement floor. Consisting of a 6½ in. x 11 in. fireclay flue lining enclosed by one course of red brick, this chimney extended up through the central part of the house. It was completely surrounded by heated space from its base to

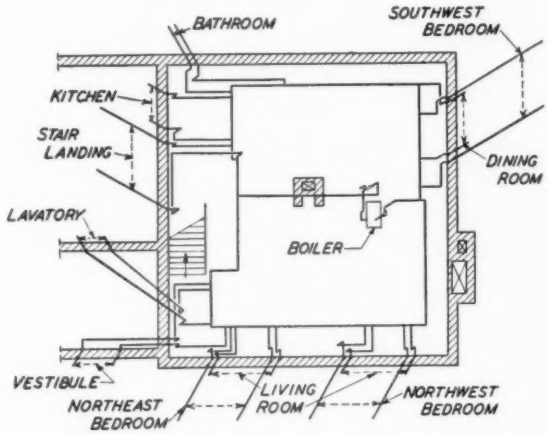


FIG. 2. BASEMENT HEATING PLAN

the level of the second story ceiling. There was no direct contact between the chimney and any part of the house framing. The plaster walls were separated from the masonry by an air space averaging about 5 in. in width. At the second story and basement ceiling levels this space was packed with mineral wool to prevent air circulation between the space around the chimney and the attic or basement. Details of the chimney construction, flue connection, and thermocouple locations are shown in Fig. 3.

TEST PROCEDURE

During the tests the flow control valve was locked open so water could circulate to the radiators by gravity during the offperiod. Operations of the burner and circulator were controlled by the room thermostat. No provision was made for heating domestic hot water. The oil burning rate of 1.1 gal per hr was the minimum obtainable with a clean fire and with a CO_2 content of the flue gases not less than 8 percent at the flue outlet of the boiler. The oil weighed 7 lb per gal and had a calorific value of 19,550 Btu per lb. To keep a constant draft on

the boiler, a draft regulator was installed in the flue pipe connecting the boiler to the chimney.

During all tests the air temperature 30 in. above the floor in the heated part of the house was maintained constant at approximately 72 F, both day and

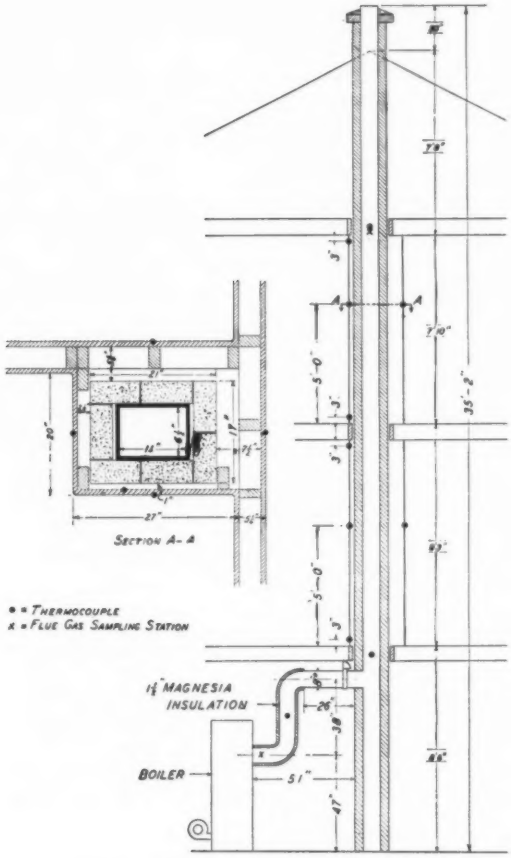


FIG. 3. CHIMNEY DETAILS

night. All windows remained closed at all times. Each test was 24 hours in length and continuous records were made of (1) the chimney gas temperatures at the levels of the attic floor, the basement ceiling, and the flue outlet of the boiler; (2) the draft in the chimney at basement ceiling level; (3) water temperature at the boiler outlet and return; and (4) outdoor air tempera-

ture. Periodically each day observations were recorded of (1) the air temperature in each room at 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling; (2) the air temperature in the basement and in the attic; (3) the surface temperatures of the plaster on the room walls surrounding the chimney; and (4) the CO_2 content of the chimney gases at the flue outlet of the boiler and at the level of the attic floor. Daily records were made of: the operating time, number of cycles and power consumption of the burner and circulator, and the weight of oil consumed.

COMPUTATIONS

Heat supplied to the house from the chimney was computed by two methods. One was based on observed flue gas temperatures and rates of flow, the other on observed temperatures of the room air and of the plaster wall surrounding the chimney.

Flue Gas Measurement Method: Heat supplied to the heated portion of the house from the chimney gases may be divided into three parts: (1) heat obtained from the dry chimney gases; (2) heat obtained from the water vapor in the chimney gases formed by the combustion of hydrogen in the fuel; and (3) heat obtained from the water vapor in the chimney gases carried in by the air for combustion.

Assuming that all heat given up by the flue gases while passing from the basement ceiling level to the attic floor level is transmitted directly to the heated section of the house, the heat obtained from the dry chimney gases during either the off- or on-period may be computed from the equation:

$$H = W_d C_p (t_b - t_c)\, \theta \quad \ldots \ldots \ldots \ldots \ldots \quad (1)$$

where

H = heat obtained from the dry chimney gases, Btu.
W_d = average rate of flow of the dry chimney gases, pounds per hour.
C_p = specific heat of chimney gases, 0.24 Btu per (pound) (Fahrenheit degree).
t_b = average temperature of the chimney gases at the basement ceiling level, Fahrenheit.
t_c = average temperature of the chimney gases at the attic floor level, Fahrenheit.
θ = time, hours.

Heat obtained from the water vapor in the chimney gases formed by the combustion of hydrogen in the fuel during the on-period may be computed from the equation:

$$H_c = 0.09\, W_o H_2 C_{ps} (t_b - t_c)\ \theta \quad \ldots \ldots \ldots \ldots \quad (2)$$

where

H_c = heat obtained from water vapor formed by the combustion of hydrogen in the fuel, Btu.
W_o = weight of oil burned, pounds per hour.
H_2 = hydrogen in the fuel, percent by weight.
C_{ps} = specific heat of the water vapor, 0.46 Btu per (pound) (Fahrenheit degree).

In addition to the water vapor resulting from the combustion of hydrogen in the fuel, about 0.005 lb of water vapor were supplied with each pound of basement air admitted to the system. Heat regained from the water vapor in this

air was less than 1.0 percent of the total heat regained and therefore was neglected in all calculations.

The rate of flow of chimney gases was not continuous and uniform, and it was not possible to calculate the rate of flow during the off-periods from the analysis of the fuel and the chimney gases as was possible for the on-periods. To use Equations 1 and 2 to calculate the amount of heat transmitted to the house from the flue gases during one day, the on-periods and the off-periods were treated separately.

Total heat transmitted to the house from the chimney gases during each on- and off-period of the burner was obtained by substituting in Equations 1 and 2 the proper values of gas flow rates and temperatures, and the analysis and weight of oil burned as obtained by test measurement.* Total heat transmitted to the house from chimney gases during any given day was obtained by taking the sum of the heat transfers computed for individual cycles occurring during that day.

Plaster Surface and Room-Air Temperature Method: A second method used in computing the amount of heat transmitted to the house from the chimney gases was based on the observed plaster surface temperature of the walls around the chimney and the room air temperatures. In making these calculations use was made of the equation:

$$H_p = A\, f_1\, (t_s - t_a)\, \theta \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

where

H_p = heat transmitted to the house from the chimney, Btu.
A = area of plaster wall exposed to rooms and enclosing chimney, square feet.
f_1 = surface conductance, Btu per (hour) (square foot) (Fahrenheit degree) = 1.65 (assumed).
t_s = average surface temperature of plaster wall exposed to rooms and enclosing chimney, Fahrenheit.
t_a = average room air temperature, Fahrenheit.
θ = time, hours.

Since there were slight differences in both the air and plaster surface temperatures for the first and second stories, the heat transfer was computed for each story separately as follows:

$$h_1 = A_1 f_1\, (t_1 - t_3)\, \theta \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

$$h_2 = A_2 f_1\, (t_2 - t_4)\, \theta \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

$$H_p = h_1 + h_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

where

h_1 and h_2 = heat transmitted from chimney to first and second story rooms respectively, Btu.
A_1 and A_2 = areas of plaster wall exposed to rooms and enclosing chimney on first and second stories respectively, square feet.
t_1 and t_2 = average surface temperatures of A_1 and A_2 respectively, Fahrenheit.
t_3 and t_4 = average room air temperatures of first and second stories respectively, Fahrenheit.

* Readers interested in a complete description of the experimental methods used to obtain the proper values to substitute in Equations 1 and 2 should refer to *University of Illinois Engineering Experiment Station Bulletin* No. 407.

The average room air temperature of the first story, t_3, was taken as the average of the air temperature readings observed each day at the 60-in. level in the living room, dining room, and kitchen. Likewise, t_4, was taken as the average of the readings observed each day of the room air temperature at the 60-in. level in the three bedrooms.

In calculating the total quantity of heat transmitted to the rooms from the chimney each day, Equations 4 and 5 may be further simplified by combining all constant terms to:

$$h_1 = c_1 (t_1 - t_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$h_2 = c_2 (t_2 - t_4) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

in which the constants c_1 and c_2 are as follows:

$$c_1 = A_1 f_1 \theta = 81 \times 1.65 \times 24 = 3208$$
$$c_2 = A_2 f_1 \theta = 72.8 \times 1.65 \times 24 = 2887$$

RESULTS

Heat Transmitted to House from Chimney Computed by the Flue Gas Temperature Method: In Fig. 4, curve 1 shows the daily heat transmitted to the house from the chimney as determined by measurements of flue gas temperatures and quantities. Test points are quite scattered as is to be expected since the coordinates selected imply that heat transfer was a function only of indoor-outdoor temperature difference. Actually, it was a function of total burner operating time, boiler water temperature and intensity of draft, all of which were affected by the amount of sunshine and average wind velocity as well as the indoor-outdoor temperature difference. Since the test observations did not make it possible to isolate effects of wind and sunshine, it was necessary to run a sufficient number of tests to permit drawing a curve representing *average* conditions.

Curve 3 of Fig. 4 represents the heat transferred to the house from the chimney as determined by measurements of flue gas temperatures and quantities, expressed in percent of the total heat loss of the house. Curve 3 was obtained by multiplying values read from curve 1 by 100 and dividing by the actual heat loss of the house for the corresponding indoor-outdoor temperature difference. This actual heat loss was obtained by measurement of all heat supplied to the Residence. A discussion of the actual heat loss of the *I=B=R* Research Residence has been published[1].

Curve 3, Fig. 4, shows that as the indoor-outdoor temperature difference increased up to about 25 deg there was a sharp increase in the useful heat obtained from the chimney when expressed in percent of the total heat loss of the house. Above an indoor-outdoor temperature difference of 25 deg this increase was less pronounced, ranging from 13.5 percent to a maximum of 14 percent. At an indoor-outdoor temperature difference of 34 deg, (representative of average winter temperatures in Urbana, Ill.) heat supplied to the house from the chimney as computed by the flue gas method amounted to 14.0 percent of the total heat loss of the house.

Heat Supplied to House from Chimney Computed by Plaster Surface Temperature Method: Useful heat obtained from the chimney each day as computed

[1] Exponent numerals refer to References.

from the observed plaster surface and room air temperatures is shown in curve 2 of Fig. 4. Test points for curve 2 are for the same days as those for curve 1 and supposedly they would have identical values if both methods of calculations were exact. Both methods of computation however, are subject to some unavoidable inaccuracies. Correlation of the test points for curve 2 in Fig. 4 is better

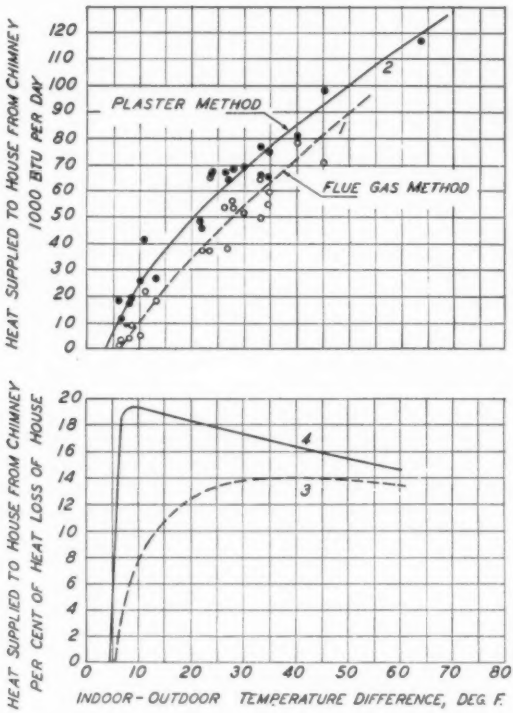


FIG. 4. HEAT SUPPLIED TO HOUSE FROM INSIDE CHIMNEY WHEN OPERATING OIL-FIRED BOILER

than that for curve 1; probably these points represent more accurately the actual heat transmission rates than do the points for curve 1 as fewer approximations are involved.

Curve 4, based on curve 2 and calculated in the same manner as curve 3, indicates that as soon as the outdoor temperature dropped low enough to require some heat in the house the useful heat obtained from the chimney expressed in percent of the total heat loss of the house rose very sharply from 0 to 19.5 percent. As the indoor-outdoor temperature difference increased, the ratio of

heat obtained from the chimney to the heat loss of the house steadily decreased to a value of about 14.7 percent at an indoor-outdoor temperature difference of 60 deg. At a temperature difference of 34 deg, useful heat obtained from the chimney as computed from observed plaster surface and ambient air temperatures, was about 17 percent of the actual heat loss of the house, as compared to the value of 14.0 percent obtained by using observed flue gas temperatures and rates of flow in computing useful heat obtained from the chimney.

Chimney Heat Transfer Efficiency: Heat transmitted to the house from the inside chimney may be expressed as a percentage of heat available in the

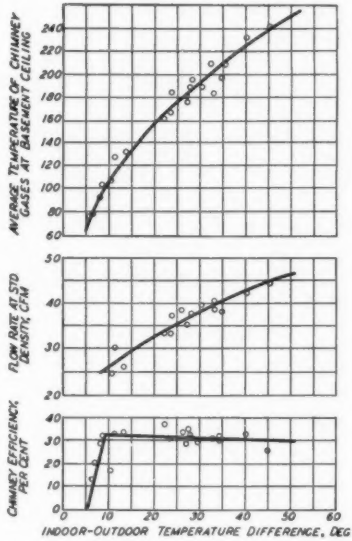


FIG. 5. CHIMNEY EFFICIENCY AND GAS FLOW RATES

chimney gases at the basement ceiling level. Since this value represents the heat transferred as related to the initial heat available, it may be designated *Chimney Heat Transfer Efficiency* or simply *Chimney Efficiency.* Chimney efficiency was determined over a range of indoor-outdoor temperature differences through observed rates of flow of chimney gases and the temperatures of these gases at the basement and second story ceiling levels. A plot of chimney efficiencies together with average flow rates and temperatures of chimney gases at the basement ceiling level is shown in Fig. 5. Chimney efficiency was fairly constant over a wide range of indoor-outdoor temperature differences and was equal to 31 percent at an indoor-outdoor temperature difference of 34 deg, which is representative of average winter temperature in Urbana, Ill.

In addition to this investigation, studies of the performance of residential chimneys have been made at the National Bureau of Standards[2] and *Battelle*

Memorial Institute[3]. Masonry chimneys were studied in all cases, although there were variations in the physical dimensions of the chimneys, the method of operation, and the type of fuel used, as shown in Fig. 6. While the total height of the chimneys differed, it should be noted that heights between temperature measuring stations used for computing the chimney efficiency were approximately equal.

From the data of tests conducted at the National Bureau of Standards and *Battelle Memorial Institute,* chimney efficiencies were computed and plotted in Fig. 6 as a function of the average temperature of chimney gases observed at the lower measuring station. Superimposed on this plot is the chimney efficiency which was obtained in the Research Home for an average temperature of

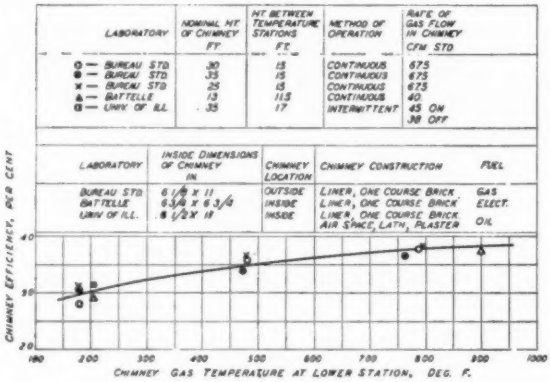

LABORATORY
NOMINAL HT. OF CHIMNEY FT.
HT BETWEEN TEMPERATURE STATIONS FT.
METHOD OF OPERATION
RATE OF GAS FLOW IN CHIMNEY CFM STD
O — BUREAU STD. 30 15 CONTINUOUS 675
● — BUREAU STD. 35 15 CONTINUOUS 675
× — BUREAU STD. 25 15 CONTINUOUS 675
△ — BATTELLE 13 11.5 CONTINUOUS 40
□ — UNIV. OF ILL. 35 17 INTERMITTENT 45 ON 30 OFF
LABORATORY
INSIDE DIMENSIONS OF CHIMNEY IN.
CHIMNEY LOCATION
CHIMNEY CONSTRUCTION
FUEL
BUREAU STD. 8 1/2 X 11 OUTSIDE LINER, ONE COURSE BRICK GAS
BATTELLE 6 3/4 X 6 3/4 INSIDE LINER, ONE COURSE BRICK ELECT.
UNIV OF ILL. 8 1/2 X 13 INSIDE LINER, ONE COURSE BRICK AIR SPACE, LATH, PLASTER OIL
CHIMNEY EFFICIENCY, PER CENT
40
30
20
100
200
300
400
500
600
700
800
900
1000
CHIMNEY GAS TEMPERATURE AT LOWER STATION, DEG. F.

FIG. 6. CHIMNEY EFFICIENCIES

chimney gases at the basement ceiling level of 208 F, obtained at an indoor-outdoor temperature difference of 34 deg (see Fig. 5). Results of the three investigations are in close agreement, even though there were differences in cross-sectional areas, total heights, and methods of operation of the various chimneys.

In residential operation, the average temperature of chimney gases will normally approximate 200 F, and should not exceed 500 F. These are not maximum temperatures, but average temperatures for both on-period and off-period operation, and furthermore, they are affected by the entrance of basement air through the draft regulator. Therefore, it would appear that a chimney efficiency of 30 to 35 percent could be expected for an inside masonry chimney having a length of travel through the heated portion of the house of approximately 15 ft.

Heat Utilization: The amount of fuel required to offset the heat loss of the house is affected by the heat supplied to the house from the inside chimney. However, heat from the chimney is not necessarily supplied to all rooms in proportion to the heat loss of those rooms. Therefore, the heat supplied by the

chimney affects both the utilization of fuel by the house and the distribution of heat within the house.

When the boiler is connected to an inside chimney, it may be assumed that the heat utilized by the house is the total heat available in the fuel burned minus the heat available in chimney gases at the upper boundary of heated space. This is equivalent to the heat transmitted to the house from the boiler and smoke pipe plus the heat transmitted from the inside chimney and may be expressed as:

$$H_L = H_f E_u + (1 - E_u) E_c H_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

or

$$H_L = H_f [E_u + (1 - E_u) E_c] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where

H_L = total heat utilized by the house, Btu per hour.
H_f = total heat available in the fuel burned, Btu per hour.
E_u = combined boiler and smoke pipe efficiency; that is, the available heat in the fuel minus the available heat in the chimney gases at the basement ceiling level divided by the available heat in the fuel burned.
E_c = chimney efficiency, decimal.

Heat transmitted to the house from chimney gases is equal to the available heat in chimney gases at the basement ceiling level multiplied by the chimney efficiency in decimal form. This may be expressed as a ratio to the heat utilized by the house as:

$$Q_c = \frac{H_f (1 - E_u) E_c}{H_f [E_u + (1 - E_u) E_c]} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

or

$$Q_c = \frac{(1 - E_u) E_c}{E_u + (1 - E_u) E_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where

Q_c = the heat obtained by the house from the chimney gases divided by the heat utilized by the house.

Equation 12 indicates that for any given value of chimney efficiency there is a direct relationship between Q_c, the heat obtained by the house from the chimney divided by the heat utilized by the house; and E_u, the combined efficiency of the boiler and smoke pipe. This relationship is shown in lower diagram of Fig. 7 for chimney efficiencies of both 30 and 35 percent.

Overall house efficiency has been defined as heat utilized by the house expressed as a percentage of the total heat available in the fuel burned. Thus, it is a measure of the heat in the fuel burned which is utilized by the house to offset heat losses. The following equation is an expression for overall house efficiency:

$$E_h = (H_L/H_f) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where

E_h = the overall house efficiency, decimal.

By substituting the equivalent of H_L from Equation 10, Equation 13 becomes

$$E_h = E_u + (1 - E_u) E_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Thus for any given chimney efficiency there is a direct relation between the overall house efficiency, E_h, and the combined efficiency of boiler and smoke pipe, E_u. This relationship is shown in top diagram of Fig. 7 for chimney efficiencies of both 30 and 35 percent.

When the chimney efficiency and either the overall house efficiency or the combined efficiency of the boiler and smoke pipe are known, Fig. 7 may be used to determine the heat supplied to the house by the chimney as a percentage of the total heat utilized by the house, Q_c. Thus Q_c is independent of the type of

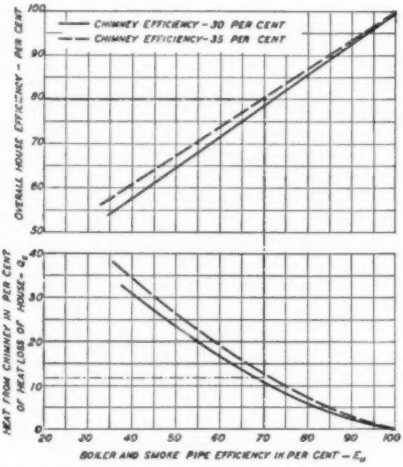

CHIMNEY EFFICIENCY - 30 PER CENT
CHIMNEY EFFICIENCY - 35 PER CENT
OVERALL HOUSE EFFICIENCY - PER CENT
HEAT FROM CHIMNEY IN PER CENT OF HEAT LOSS OF HOUSE - Q_c
BOILER AND SMOKE PIPE EFFICIENCY IN PER CENT - E_u

FIG. 7. RELATION BETWEEN HEAT FROM CHIMNEY AND OPERATING EFFICIENCIES

construction of the house, the type of fuel burned, and the indoor-outdoor temperature difference.

The range of chimney efficiencies for an inside masonry chimney in a two-story house will normally be between the values of 30 and 35 percent. Results of heating research[4,5] have shown that overall house efficiencies are generally about 80 percent or less. With an overall house efficiency of 80 percent and a chimney efficiency of 33 percent, Fig. 7 shows that the combined efficiency of the boiler and smoke pipe must be 70 percent and heat supplied to the house by the chimney in percent of the total heat utilized by the house, Q_c, must be 12 percent. For a two-story house, this averages 1.5 percent of the total heat utilized by the house for each side of the chimney exposed to a room. At design outdoor temperatures, the total heat utilized by the house is assumed to be equal to the calculated heat loss of the house. Therefore, to maintain uniform temperatures in the various rooms, heat to be supplied by the installed radiation in a room should be reduced by an amount equal to 1.5 percent of the calculated heat loss of the *house* for each side of the chimney exposed to that room. For

example, in a residence with a calculated heat loss of 50,000 Btu per hour, the reduction of installed radiation in a room to which two sides of the chimney are exposed would be equal to 1500 Btu per hour. If the overall house efficiency decreases below 80 percent, heat obtained from the chimney in percent of the heat loss of the house will increase. At an overall house efficiency of 70 percent, Fig. 7 shows that the heat obtained from the chimney will be about 20 percent of the heat loss of the house. Radiation installed according to the foregoing procedure should be adequate.

Correction for Chimney Heat Applied to I=B=R Research Home: Column 3 of Table 1 gives the total calculated heat loss of each room of the *I=B=R* Research Home at the design indoor-outdoor temperature difference of 80 deg. Conven-

TABLE 1—DATA ON *I=B=R* RESEARCH HOMES ILLUSTRATING USE OF RADIATION SELECTION PROCEDURE

ROOM	NO. OF SIDES OF CHIMNEY EXPOSED TO ROOM	CALCULATED HEAT LOSS BTU PER HR	HEAT OBTAINED FROM CHIMNEY BTU PER HR	NET HEAT TO BE SUPPLIED BY RADIATION BTU PER HR	REDUCTION IN RADIATION REQUIRED PERCENT
(1)	(2)	(3)	(4)	(5)	(6)
Living Room	1	5749	650	5099	11.3
Dining Room	2	8742	1300	7442	14.9
Kitchen	1	3199	650	2549	20.3
Lavatory		1484		1484	
Vestibule		4848		4848	
Vestibule Closets					
N.E. Bedroom		4393		4393	
N.W. Bedroom	1	4944	650	4294	13.1
S.W. Bedroom	2	5250	1300	3950	24.8
Bath		2606		2606	
Stair	1	2155	650	1505	30.2
Closets					
Total		43,370	5,200	38,170	12.0

tional practice in the design of a steam or hot water heating system is to provide sufficient radiation in each room to offset this calculated design heat loss. Column 2 indicates the number of sides of the inside chimney which are exposed to the different rooms. Column 4 is the allowance for heat obtained from the chimney, (taken as 1.5 percent of the calculated heat loss of the house for each side of the chimney exposed to the room), and column 5 represents the net heat to be provided by radiation after making allowances for heat obtained from the chimney.

Using the conventional procedure the radiation would be selected in accordance with the values in column 3 of Table 1. By the procedure suggested in this paper the radiation would be selected on the basis of the values in column 5. Column 6 shows that in the Research Home heat obtained from the inside chimney was sufficient to take care of from 11 to 30 percent of the normal heat requirements of the rooms to which the sides of the chimney were exposed.

The first heating system installed in the Research Home[5] had an amount of radiation installed in each room just sufficient to offset the calculated heat loss

of the room (column 3, Table 1). To obtain proper balance of room temperatures it was found necessary to reduce the radiation in the dining room and southwest bedroom by 35 and 25 percent, respectively, while the radiation in the living room had to be increased by about 9 percent.

In the southwest bedroom the reduction made in the amount of installed radiation corresponded very closely to the reduction indicated by the design procedure suggested in this paper. In the dining room the actual reduction was even greater than that indicated by the estimated chimney allowance. The boiler was in the basement directly under the dining room and temperatures taken on either side of the dining room floor indicated that additional heat was supplied to this room through the floor.

Only in the living room was the actual change in radiation contrary to that indicated by the proposed procedure. Observations in the Research Home have indicated that the radiation installed in the vestibule was not adequate to keep the vestibule as warm as the remainder of the house. As a result, cool air circulated through the open doorway from the vestibule into the living room, increasing the heat requirements of the latter. Had sufficient radiation been installed in the vestibule it is unlikely that additional radiation would have been required in the living room.

Thus, actual performance has indicated that conventional procedures for the selection of radiation have resulted in too much radiation in those rooms of the Research Home to which two sides of the chimney were exposed. Furthermore, had the procedure suggested in this paper been used the room temperature balance would have been improved in every room with the possible exception of the living room. In this room the unbalanced condition resulted largely from lack of sufficient radiation in the adjoining vestibule, rather than from heat obtained from the chimney.

REFERENCES

1. Performance of a Hot-Water Heating System in the *I=B=R* Research Home at the University of Illinois, by A. P. Kratz, W. S. Harris, M. K. Fahnestock, and R. J. Martin (University of Illinois, Engineering Experiment Station Bulletin No. 349, Section 26).

2. Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach and J. T. Duck (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 351).

3. Data taken from unpublished progress report on the Performance of Residential Chimneys, by R. B. Engdahl, P. C. Vyff, and F. B. Terrill (May 13, 1946).

4. Investigation of Oil-Fired Forced-Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo (University of Illinois, Engineering Experiment Station Bulletin No. 318, pp. 39-44).

5. See Reference 1, pp. 57-62.

DISCUSSION

P. R. ACHENBACH, Washington, D. C.: The Kacheloven is an interesting extension of the idea of using heat transferred to living space from an inside chimney. The Kacheloven used considerably in Europe, has been installed in a few public housing developments in Pittsburgh. The combustion chamber of the Kacheloven is built into a large chimney and almost all heating of the house takes place through the chimney walls. Gases are carried up through a large chimney liner, baffled on the inside

to promote better heat transfer. Heat transferred from the exterior of the chimney is used to heat the house by radiation and convection. Some people appear to favor these heating units, but others dislike the lag in the transfer of heat through such heavy masonry construction. Performance of the Kacheloven represents one of the limits of the curves in Fig. 7 of this paper for which the efficiency of the boiler and smokepipe, E, was near zero and the percentage of the total heat loss of the house obtained from the chimney, Q_c, was nearly 100.

Care should be exercised in using the overall house efficiency as a measure of the heat available for providing comfort in living spaces. It cannot be assumed that because a house receives heat from various sources, that the heat is necessarily distributed in exactly the right amounts to produce comfort in the occupied spaces. A heating system installed in a basement without radiators or ducts above the floor level might have a satisfactory overall house heating efficiency, but it would not adequately warm the living space in most houses.

This paper suggests one other word of caution to heating engineers. We should not be entirely satisfied with present methods of computing heat losses of buildings and selecting heating plant capacities, because of such unknowns as this paper describes. It points up, rather, that further investigation of lesser-known heat sources not taken into account in present calculations, would be advantageous.

R. K. THULMAN, Washington, D. C.: This is a contribution to the calculation of vagrant heat losses in a house. These sometimes upset results, and perhaps produce a situation in the performance of the heating system for which the designer was unprepared.

Chimney research that the Housing Agency has sponsored at the National Bureau of Standards and, at the *Battelle Memorial Institute*, has been directed toward discovery of a more efficient chimney.

Chimney efficiency is a proper measure of the function of the chimney itself as a draft producer. Draft production in a chimney is improved by keeping as much of the heat in the chimney as possible and conserving as much heat as possible. It therefore seems unfortunate to use the term *chimney efficiency* for something that is designed against.

E. K. CAMPBELL, Kansas City, Mo.: The real purpose of a paper of this sort is to point up the inefficiency and consequent heat loss of the boiler or the furnace. That loss was greater years ago when equipment was mostly hand-fired with coal.

In a number of cases we have dealt with, economizers were installed on the outlet of cast-iron boilers to save heat. In one case, gas burners were installed in old boilers, together with the economizers. This was a church installation, and a small fan blowing air over the economizer heated the basement of the auditorium and helped to heat the main auditorium.

An amusing complaint was registered that the economizer was stopped up. Investigation disclosed that the economizer was clean but it had retained, or saved, so much heat that there was not enough heat to furnish chimney draft. It was a large chimney, and the condition was corrected by reducing the size of the flue.

An important consideration in the design of any heating plant is safety. A frequent cause of trouble is lack of capacity. Therefore, no heating system should be designed so closely that losses due to leakage and other causes will endanger the safety of the system.

AUTHORS' CLOSURE (R. J. Martin): The authors appreciate the discussions of this paper since they have brought out several important points.

Mr. Achenbach has indicated that the heat received by a house from various sources is not necessarily distributed in such a way as to produce comfort. This is exactly correct and is one of the reasons that we should have a better knowledge of vagrant heat sources and their effect on balancing the heating system for maximum comfort

and economy. Tests at the *I=B=R* Research Home have shown that only about 50 percent of the heat utilized by the house was supplied by radiators while the other 50 percent of the heat utilized was supplied from such sources as the chimney, piping, boiler jacket and smoke pipe, people, lights, etc. This paper has shown that from 12 to 15 percent of the total heat required by the house was from the chimney and also it has shown how this heat source can be readily determined and provided for in the design of the system. At present day costs for materials and labor, this possible savings in heating system costs would seem worthwhile—particularly if the resulting system was more nearly balanced to the heating requirements.

Mr. Thulman has indicated that a possible confusion might arise from our use of the term *chimney efficiency*. It should be pointed out that the complete phrase is *chimney heat transfer efficiency*. In our tests we did not in any way attempt to change the natural draft produced by the chimney, nor was the chimney designed or constructed to provide more heat to the house than normal. This was an inside masonry chimney which was constructed in the usual way and, incidentally, it produced more draft than was required by the boiler.

It is not clear what is meant by *designing against chimney efficiency*. I presume Mr. Thulman means that we should design a chimney to have a low heat transfer efficiency in order that a high draft will be produced. This is not necessarily the case since it is temperature of the column of hot flue gases in the chimney that produces the draft and not whether the heat is transferred to the house or lost out of the top of the chimney to the surroundings. A chimney located on the outside wall of a house, for example, generally has lower draft producing capacity and also has a lower heat transfer efficiency than an inside chimney of the same construction.

Mr. Campbell has mentioned the low efficiencies of boilers. This is not too material to our problem as long as the house utilizes a large portion of the heat supplied to the boiler. Tests in several different research homes at the University of Illinois have shown overall house efficiencies of 80 percent and higher. I do not believe that this could be called a low utilization efficiency, in view of the *required* loss to remove the products of combustion from the system.

It is good engineering practice to install a heating system that is the *proper* size, not one which has been oversized by adding factors of safety which in fact do not add at all to the safe operation of the system. If we are to have the most satisfactory operation of a heating system, it must be designed to match the heating load and not be oversized to the point where proper control of the system is impaired.

The Society and the heating industry as a whole have for years been studying the problems of heating loads and heat utilization. We need to know even more about this subject, and also to put into practice the information which is now available.

1468

No. 1468

FUEL CONSUMPTION ANALYSIS FOR MULTI-FAMILY HOUSING PROJECTS

By RUTCHER SKAGERBERG* AND J. E. PHIFER**, WASHINGTON, D. C.

VARIOUS means of statistical evaluation of the management operations of public housing projects have been used during the period of rapid growth of such projects. For heating operations it was apparent that the quantities of fuel used might best serve that purpose. The data were collected only where management supplied the fuel because it was only from such projects that fuel consumption data could be obtained.

Now that 11 years have elapsed it seems timely to analyze the data to see if there are any indications or trends that might be significant and useful to planners, designers and operators. That is the purpose of this paper.

TECHNIQUES AND DEFINITIONS

After collecting the fuel consumption data the tons of coal, gallons of oil and cubic feet of gas were converted to equivalent heat input in Btu.

The unit Btu per room per year was selected as a useful and easily obtained unit for comparison purpose. Cubic foot volume and the square feet of floor area were considered in place of the room basis but were dropped because it was more practicable for the purposes in view to use the room as a basis since the rooms were reasonably standard and the room count could readily be obtained and agreed upon.

Space heating fuel could not be compared on a Btu-per-room basis without considering the climate and consequently the unit was changed to Btu-per-room-per-degree-day. It seemed, however, that fuel usage might not be proportional

* Director, Operations Engineering Branch, Housing and Home Finance Agency. Member of A.S.H.V.E.

** Assistant to Director, Operations Engineering Branch, Housing and Home Finance Agency.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

to degree-days in different climates and therefore, the country was divided into six zones having degree-days as follows:

ZONE	DEGREE-DAYS
I	Less than 1500
II	1500 to 3499
III	3500 to 4499
IV	4500 to 5499
V	5500 to 6499
VI	6500 and over

Zone I is not involved in these analyses as no space heating is used in it. Since few project-operated heating plants are used in the warmest climates, Zone II involves a range of 2000 degree-days whereas each of the other zones covers only a 1000 degree-day range.

While in the coldest zones space heating constitutes the largest portion of the heating plant load, the fuel used for domestic hot water, underground line losses and boiler losses is a considerable amount. For these latter usages which were not considered to be variable with weather the degree-day unit did not apply. Therefore, it was necessary to separate space fuel requirements from these other items and, consequently, three fuel quantities were considered:

1. *Total fuel,* consisting of the entire annual fuel consumption expressed in Btu-per-room-per-year.

2. *Space fuel,* consisting of the fuel used for space heating and expressed in two units: *Btu-per-room-per-year* and *Btu-per-room-per-degree-day.*

3. *Other fuel,* consisting of fuel consumed for domestic hot water, laundry purposes, underground pipe line losses, and boiler house losses.

Since the projects report only the total quantities of fuel used each month it was necessary to find a simple and practical method of determining the *space fuel* and *other fuel* amounts. This was done by taking the *total fuel* for the months of July and August as representing *other fuel* since no *space fuel* was used then. Assuming that *other fuel* was the same for each month, the combined July and August quantities multiplied by six provided the annual value for *other fuel* which was subtracted from the annual *total fuel* to obtain a value for annual *space fuel.*

Three questions arose as to the accuracy of the assumption relative to the uniformity of the *other fuel* usage throughout the year:

1. There was the question of different quantities of domestic hot water used in different seasons. This was checked over a two-year period by metering the hot water for 10,000 families. There was noted no appreciable differences between seasons although there was a difference between projects and between families within projects ranging from 33 to 78 gal per day and averaging 56.

2. The effect of colder incoming water in winter than in summer seemed to require some adjustment. At the same time the improved efficiency of boilers during heavy-load periods *vs.* light summer load conditions was considered important. Investigations revealed that one of these factors almost offsets the other. In fact the difference was so slight when comparing final results that it was not worth the great mass of computations required. Therefore, no corrections are made for these two factors.

3. Underground line losses were also investigated. While the tests performed were not conclusive it was decided that the thermal lag of the earth had a tendency to even out this matter. As an example the straw-hatted workmen in Saskatchewan in Canada are reported to thaw out underground water lines in June. Also in Washington, D. C., the maximum underground temperature is achieved in October and the minimum in April. In other words it is conceivable that underground losses may be higher in July than in November or December.

In view of considerations 1 to 3 it is believed that the practical method employed in computing space heat fuel requirements is sufficiently precise for the purposes of evaluating performances and for providing data for the analyses in this paper.

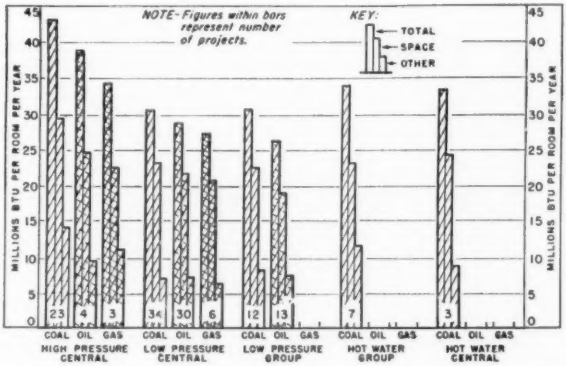


FIG. 1. EFFECTS OF PLANT AND FUEL TYPES ON TOTAL, SPACE AND OTHER FUELS

RESULTS

Ten figures are presented, each of which was prepared to show the effect on fuel consumption of one or more factors having influence and being subject to change by the planner, designer, or manager. The comments and observations applying to each chart represent the authors' interpretation of data obtained in the field.

The term, *Btu-per-room-per-degree-day*, is referred to as the *Fuel Burning Rate* or simply the *Fuel Rate* as a simplification. *Space fuel* is used to denote quantity of fuel for space heating.

EFFECTS OF PLANT AND FUEL TYPES ON TOTAL, SPACE AND OTHER FUELS

Fig. 1 is based on data for 135 projects, covers all zones, and represents a cross section of the entire country. Comparisons are based upon the Btu-per-room-per-year.

Observations

1. Low pressure central or group plants are more economical than high pressure central plants when burning coal.

2. Oil is a more economical fuel than coal in low pressure central or group plants.

3. Group plants with shorter and smaller underground lines do not, as might be expected, show lower quantities of *other fuel* than low pressure central plants. Lower summer boiler efficiency probably is responsible.

EFFECTS OF PLANT AND FUEL TYPES ON SPACE FUEL (FIG. 2)

Space fuel consumption can best be analyzed on a degree-day basis although, as will be shown later, comparisons between localities with widely different average temperatures will necessitate the use of a correction factor (see Fig. 5). This chart, based upon 135 projects, shows a country-wide comparison of space

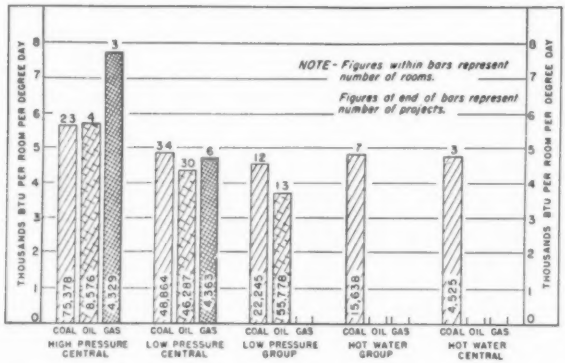


FIG. 2. EFFECTS OF PLANT AND FUEL TYPES ON SPACE FUEL

fuel expressed in Btu-per-room-per-degree-day, and is a more realistic comparison of performance than Fig. 1.

Observations

1. Low pressure plants show a lower fuel consumption than high pressure plants.

2. For low pressure plants, oil shows a preference over coal. Factors which contribute to this difference are the more consistent quality of fuel when using oil, automatic firing, and less reliance on operators.

3. Hot water group plants compare favorably with low pressure central plants using coal.

SPACE FUEL STUDIES (COMPARISON OF HEATING CONTROL SYSTEMS AND OF FUELS)

The data upon which Fig. 3 is based represent up to 11 years of operation of 92 projects in the three coldest Zones, IV, V and VI.

Each bar represents a project and is classified as to type of fuel—coal, oil, or purchased steam, and is arranged on the chart with others of the same fuel. Within each fuel group the projects are again grouped by types of heat control,

and the order in which the bars appear corresponds to the number of annual degree-days of the locality, the lowest being at the left.

Two types of heat control systems are used with steam heating in public low-rent housing projects: *on-and-off* and *continuous flow,* more accurately described as *open-and-shut* and *modulating.* A third type is the *hot water temperature* type and is used with hot water systems.

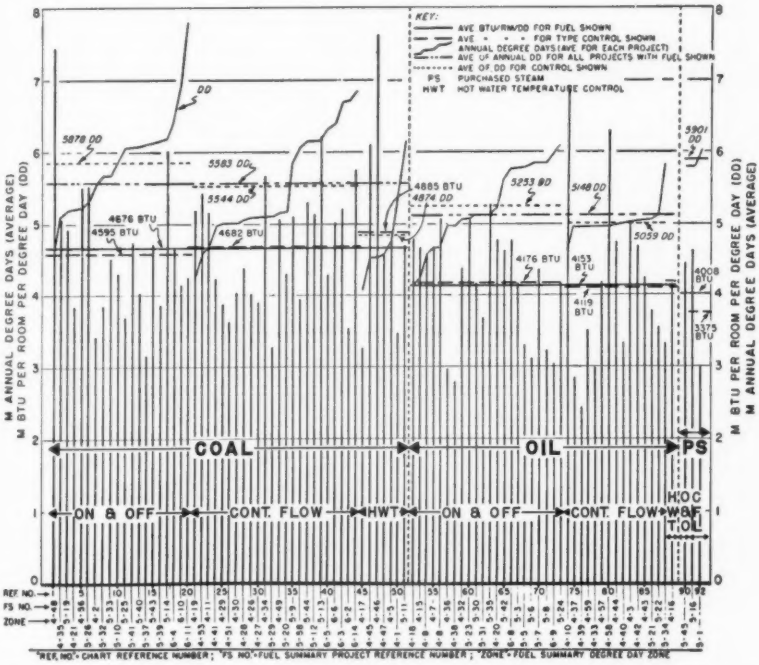


FIG. 3. SPACE FUEL STUDIES (COMPARISON OF HEATING CONTROL SYSTEMS AND OF FUELS)

While these control systems have been called *zone controls,* they are not actually zoned as to exposure. Rather, they regulate the flow of steam to entire buildings or groups of buildings with no regard to exposure of the different parts or sides of those buildings. This regulation is obtained through instruments that detect weather changes such as temperature, wind and solar intensity.

Observations

1. Oil shows 11 percent lower consumption than coal: 4153 *vs.* 4676 Btu-per-room-per-degree-day respectively. Purchased steam with unit consumption of 4008 Btu compares favorably with oil.

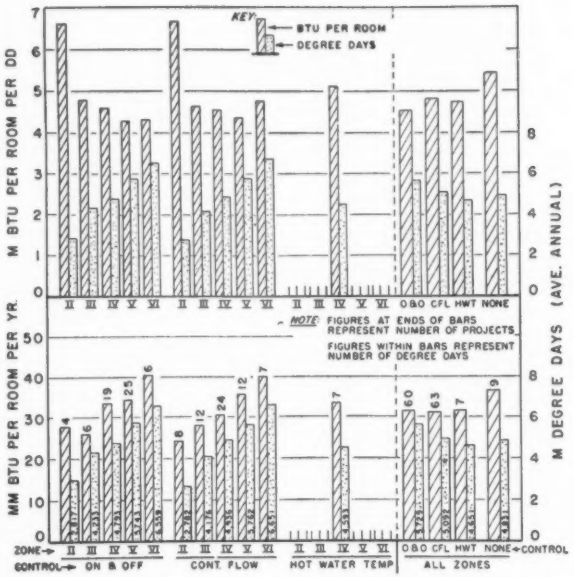

KEY
BTU PER ROOM
DEGREE DAYS
M BTU PER ROOM PER DD
M DEGREE DAYS (AVE. ANNUAL)
MM BTU PER ROOM PER YR.
O&O CFL HWT NONE
NOTE: FIGURES AT ENDS OF BARS REPRESENT NUMBER OF PROJECTS
FIGURES WITHIN BARS REPRESENT NUMBER OF DEGREE DAYS
ZONE→ II III IV V VI II III IV V VI II III IV V VI O&O CFL HWT NONE→CONTROL
CONTROL→ ON & OFF CONT. FLOW HOT WATER TEMP ALL ZONES

FIG. 4. EFFECTS OF HEAT CONTROL OPERATION

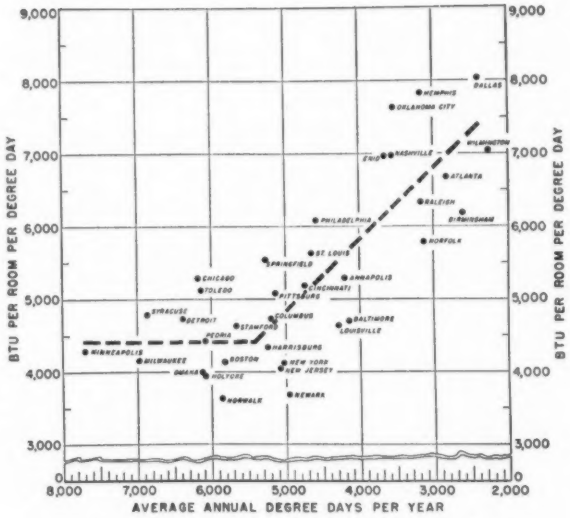


FIG. 5. EFFECTS OF CLIMATE ON FUEL RATE

2. The oil installations are in lower average degree-day localities than coal and purchased steam: 5148 *vs.* 5583 for coal and 5901 degree-days for steam. As will be shown later, the performance (per degree-day) generally improves with lower outdoor temperatures, *i.e.*, greater degree-day values.

3. The differences in average performance of controls of different types appear negligible.

4. A tendency is shown toward higher fuel consumption per degree-day in milder climates. As will be shown later, this is much more pronounced in more southerly zones which are not included in Fig. 3.

5. Performances of different projects vary quite widely, although of the coal plants 37 percent lie within ± 10 percent of the average, and 70 percent within 20 percent. Of the oil plants, the values are 25 percent and 65 percent respectively.

Differences in project performance can be attributed to such factors as: (1) management control of overheating; (2) conditioning of equipment; (3) skill and interest of operators; (4) wind and other weather effects aside from temperatures; (5) policy on such matters as night shut-down; (6) condition and kind of hot water generating equipment; and (7) plumbing. The last items may show high values for *other fuel* and result in exceptionally low values of *space fuel* or vice versa.

EFFECTS OF HEAT CONTROL OPERATION

Fig. 4 is based upon space heat data for 130 projects. The lower half is expressed in Btu-per-room-per-year; the upper half in Btu-per-room-per-degree-day. The performance of any project may be evaluated by comparing its fuel rate with others in the same degree-day zone and with the average fuel rate of that zone.

Observations

1. The all-zones comparison (right side of Fig. 4) shows the different types of controls to be about equal in performance. The slight advantage of the *on-and-off* type in the fuel rate comparison (upper right) may be accounted for by the fact that they predominate in colder climates.

2. Controlled heating systems show less fuel consumption than uncontrolled systems. The upper chart indicates a saving of 16 percent and the lower, 13 percent.

3. Here again is illustrated the fact that while with increasingly colder temperatures the annual fuel consumption increases, the rate of consumption per degree-day decreases.

EFFECTS OF CLIMATE ON FUEL RATE

Since the degree-day is generally used as a measure of heat demand and is easily obtained, it was determined a few years ago that a check should be made as to its accuracy as such a measure. There was considerable evidence that fuel rate was greater in milder weather.

Fig. 5 is presented to illustrate this point and to suggest a possible formula that might be generally applied. This represents 35 cities whose annual degree-days range from 2300 to 7800. It shows the fuel rate for all types of heating plants and fuels.

Observations

1. In coldest climates the annual fuel rate is constant for a considerable range of annual degree-day values. In moving toward milder climates a point is reached at which the rate begins to climb.

2. The scattered arrangement of points on the chart may suggest several trend lines either curved or made up of combinations of curved and straight lines. It is apparent that a rather definite pattern exists. By averaging groups from left to right there is developed a horizontal-line constant rate at 4400 Btu to the 5300 degree-day ordinate, then an upward slope of 45 deg for milder weather. The rate increases by 1 Btu for each rise of 1 degree-day.

A line of this kind might be used as an approximate criterion for predicting fuel consumption and evaluating performance. It would obviously be more accurate than

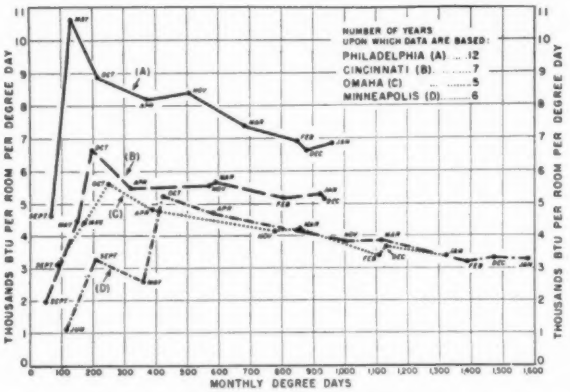

NUMBER OF YEARS UPON WHICH DATA ARE BASED:
PHILADELPHIA (A) ... 12
CINCINNATI (B) ... 7
OMAHA (C) ... 5
MINNEAPOLIS (D) ... 6
THOUSANDS BTU PER ROOM PER DEGREE DAY
MONTHLY DEGREE DAYS

FIG. 6. VARIATIONS OF FUEL RATE BETWEEN DIFFERENT HEATING MONTHS

an assumption of a fixed rate per degree-day for all localities. A study of each project is needed to obtain its own individual measuring stick or norm.

VARIATIONS OF FUEL RATE BETWEEN DIFFERENT HEATING MONTHS (FIG. 6)

In connection with the variations which were illustrated in Fig. 5, it is also important to note the differences in consumption during mild and cold weather in any one locality. If a degree-day factor is to be used as a guide for determining fuel demand, it is necessary to know what fuel rate to expect for each month. This is done by obtaining the average for each month over a number of years. Fig. 6 shows the variations in fuel rate for four localities—Philadelphia, Cincinnati, Omaha and Minneapolis, where project daily fuel consumptions were accurately measured and recorded.

Observations

1. There is striking similarity of fuel rates for the four projects.

2. From October through January, the rate decreases about 1500 Btu in Cincinnati and 2400 Btu in Omaha, the average rate of decrease being about 2000 Btu.

3. During the three coldest months (December through February) the fuel rates are almost equal for any one project.

4. September has a low fuel rate. This is attributed to the fact that degree-days are established from cold nights and since days are warm no fuel is then needed. Many projects practice night shut down and obtain a low fuel consumption per degree-day.

5. For Omaha and Minneapolis, the fuel rate in May is lower than in October; in Minneapolis it is also lower than in September. It appears that late autumn weather has chilling experience after the summer's heat whereas the weather in May has a warming effect on people who have been acclimated to cold winters. In other words, in October people have not yet donned heavier clothes and in May they have not yet removed them. Therefore, the tendency is to require more heat in October and less in May.

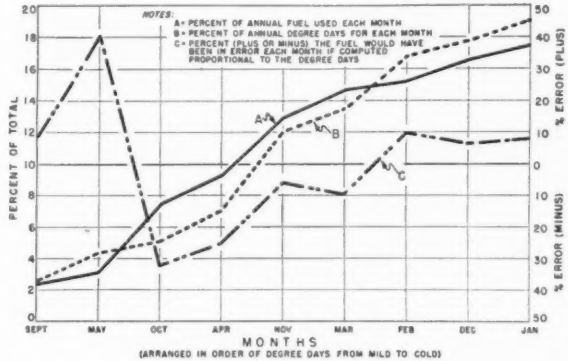


FIG. 7. ERROR IN ESTIMATING FUEL REQUIREMENTS BY PROPORTIONAL DEGREE-DAY METHOD

6. If the degree-day is to be used as a factor in evaluating fuel rate, it is necessary to determine a norm for each month or for some other short period. The annual rate cannot be applied to monthly or weekly periods.

7. For accurate budgeting it is necessary to develop a separate criterion for each project.

ERROR IN ESTIMATING FUEL REQUIREMENTS BY PROPORTIONAL DEGREE-DAY METHOD

Fig. 7 shows a percentage presentation of the monthly fuel consumption for the Minneapolis project shown in Fig. 6. The error in estimating fuel consumption by assuming the rate to be proportional to the degree-days, as previously discussed, is also illustrated by line C referred to the (% error) scale on the right.

Observations

1. The important presentation is shown by line C. It is obvious that monthly consumption is not proportional to the number of degree-days therein, for when that method of estimating is employed, differences from actual consumption range from 40 percent *over* for May to 32 percent *under* for October. However, the errors of over-

estimating in December, January and February are offset by errors of underestimating in March and November so the annual fuel estimate may not be greatly affected by these monthly differences.

2. The relationship between lines A and B (percent of annual fuel and percent of annual degree-days) illustrates again the fact that the fuel rate is less during cold than during mild weather.

INFLUENCE OF DENSITY ON FUEL CONSUMPTION

The density of a project is defined as the number of dwelling units per acre of land under management control, omitting vacant and unused land and large play fields. Fig. 8 shows the fuel rates of 21 coal-burning low-pressure plants.

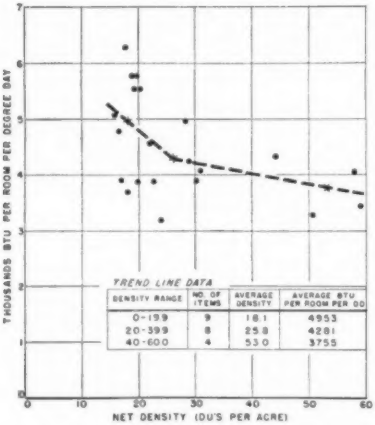

THOUSANDS BTU PER ROOM PER DEGREE DAY
NET DENSITY (DU'S PER ACRE)
TREND LINE DATA
DENSITY RANGE | NO. OF ITEMS | AVERAGE DENSITY | AVERAGE BTU PER ROOM PER DD
0-199 | 9 | 181 | 4953
200-399 | 8 | 258 | 4281
400-600 | 4 | 530 | 3755

FIG. 8. INFLUENCE OF DENSITY OF FUEL CONSUMPTION

Observations

1. While the points are quite scattered, there is a definite trend toward more economical heating in greater density projects. Other factors, such as project size and geographical locations, exert some influence here. It is necessary, therefore, to analyze each situation with its own conditions in mind.

EFFECTS OF PROJECT SIZE ON FUEL CONSUMPTION

Fig. 9 gives consumption data for 69 projects and shows in a general way the approximate influence of project size. Only central plants were included in this study to avoid the influence of plant type. Coal and oil as well as high and low pressure systems are identified by different symbols. Two categories of size were chosen, *i.e.*, below and above 2,000 rooms. In view of the number of plants involved of each type and the scattered arrangement of points, the relative

fuel rates are shown by straight lines as the most practical criterion. This chart is an endeavor to provide a rough guide for the selection of systems.

Observations

1. The average of all large-project (2,000-5,000 rooms) fuel rates is 12 percent lower than that of the small projects.

2. The average of fuel rates for all large projects with high pressure plants burning coal or oil is about 12 percent under the similar average for small projects.

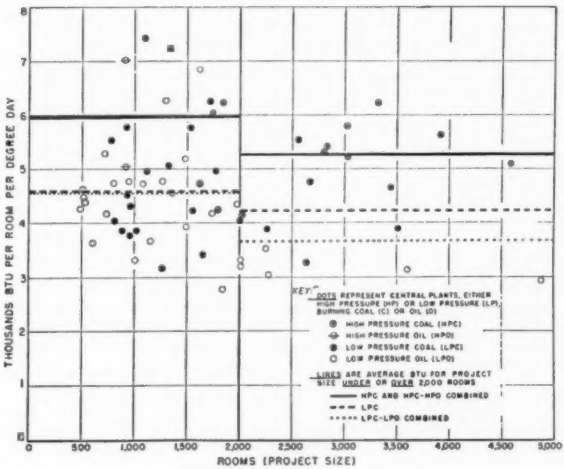

THOUSANDS BTU PER ROOM PER DEGREE DAY
ROOMS (PROJECT SIZE)
KEY
DOTS REPRESENT CENTRAL PLANTS, EITHER HIGH PRESSURE (HP) OR LOW PRESSURE (LP), BURNING COAL (C) OR OIL (O)
HIGH PRESSURE COAL (HPC)
HIGH PRESSURE OIL (HPO)
LOW PRESSURE COAL (LPC)
LOW PRESSURE OIL (LPO)
LINES ARE AVERAGE BTU FOR PROJECT SIZE UNDER OR OVER 2,000 ROOMS
HPC AND HPC-HPO COMBINED
LPC
LPC-LPO COMBINED

FIG. 9. EFFECTS OF PROJECT SIZE ON FUEL CONSUMPTION

3. The average of fuel rates for all large projects with low pressure plants burning coal is about 9 percent under the similar average for small projects.

4. The average of fuel rates for all large projects with low pressure plants burning coal or oil combined is about 20 percent under the similar average for small projects.

COMPARISON OF FUEL CONSUMPTION FOR DOMESTIC HOT WATER AND UNDERGROUND STEAM LINE LOSSES

This study was made to provide some idea of the relative amounts of fuel chargeable to line losses, domestic water heating and boiler losses (see Fig. 10). The assumption was made that these items remain relatively constant throughout the year.

Since space fuel varies approximately directly with degree-days, the percentage of fuel represented by other items will naturally vary inversely with degree-days. The basic data were obtained from a project at Minneapolis.

Observations

1. Fig. 10 is fairly self-explanatory. It should be observed, however, that the amount of fuel lost through transmission lines closely approximates that used for providing hot water. Line losses vary from 7½ percent below that used for domestic hot water during the month of June to only 1½ percent below during the months of December and January. It should be noted that the above comments are based upon operating data for one typical project, not the averages of a number of projects.

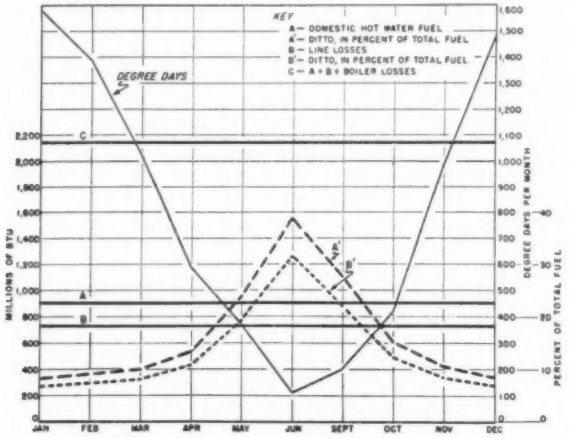


FIG. 10. COMPARISONS OF FUEL CONSUMPTIONS FOR DOMESTIC HOT WATER AND UNDERGROUND STEAM LINE LOSSES

CONCLUSIONS AND INDICATIONS

In view of the numerous situations that make for differences in heating operations the number of projects in each possible category is rather small. Furthermore 11 years of operation does not constitute enough experience to furnish conclusive evidence or good norms for some aspects of the analysis. Therefore, it is somewhat premature to draw conclusions from some of these data. However, there are a number of indications that may at a later date be resolved into definite conclusions. Other indications are sufficiently pronounced at this time so that they may be regarded as conclusive.

The following *conclusions* are offered:

1. Low pressure steam plants are less wasteful of fuel than high pressure plants.
2. Underground pipeline losses constitute a relatively large part of the heating load of a central system and deserve consideration particularly from the viewpoint of central plant adaptability to projects of low density with long underground runs.
3. High density projects, compared to low density projects, consume less fuel with central plants and where they do not adapt themselves to individual heating should be considered for central plant heating.

4. Automatic control of the distribution of steam to dwellings in housing projects can be counted upon to effect fuel savings as compared to operation without such controls. A saving of 15 percent is conservative with average operation. Greater returns can be expected where operators are fully instructed, cooperative and under good management.

5. In any multiple-family housing project with a central or group heating plant the fuel consumption per degree-day can be considered constant during December, January and February. For milder weather that value must be increased. Similarly the value is different in different localities particularly when the annual fuel consumption is compared for northern and southern locations.

The following *indications* are apparent:

1. Oil lends itself to lower fuel usage than coal. This is influenced by the ease with which firing control is accomplished, the more uniform quality of fuel, and the reduction of reliance on the human factors.

2. Hot water central or group heating plants compare favorably in fuel economy with low pressure steam plants.

3. The difference, on the average, in performance of different types of heat control systems appears negligible. A simple, easily understood, rugged, less expensive means of automatic control may be more effective economically than a more perfect but more complicated system that may easily become inoperative.

4. A fuel consumption formula based on degree-days may be predicated on a uniform fuel rate (Btu-per-degree-day) during the three coldest months or annually in localities with more than 5,000 annual degree-days. For milder months or localities the rate increases and a correction factor is needed.

In view of the numerous factors that influence fuel usage, it will be necessary to make a more detailed study of each plant to find reasons for its departure from what might be considered a *norm* for its type. Such studies are quite involved and beyond the scope of this endeavor. A recently adopted policy for making periodic engineering surveys of properties is bringing forth information that may be of value for this purpose.

A 15-year summary is now planned which should be an improvement over the last summary, both in method of presentation and coverage. It is expected that this summary will be issued during 1954 and that the data contained therein will be more significant in pointing toward important conclusions.

DISCUSSION

M. J. REED, New York, N. Y.: I have no particular criticism to make of this very constructive paper, which fills in a big gap in our quantitative information on the subject, but I should like to ask two questions:

1. In Fig. 6 where four curves are presented for four different localities; I am wondering if each curve covers one housing project only, and if so, what type of fuel was used? If each curve covers more than one housing project, are there any of these curves which are for only one type of fuel? For instance, if there happened to be three housing projects, might they all burn coal or all burn oil?

2. Can the night setback practice be identified for these curves, since most of these housing units shut down altogether for the night, although some of them may set back? Can we identify any of these curves as invariably total shutdown or invariably night setbacks of a certain number of degrees?

AUTHORS' CLOSURE (Rutcher Skagerberg): Each curve represents a central heating plant. Each plant serves one project excepting in the case of Cincinnati where there are three developments but two of them are additions to the original so that all three constitute one development.

Coal is burned as follows:

Philadelphia: West Virginia bituminous
Minneapolis: Southern Illinois (Delta Mines)
Omaha: Missouri mine run
Cincinnati: West Virginia, Dist. No. 8, Nut and Slack

The plant operation practices relative to shut down are as follows:

Philadelphia: Boilers are banked at 10 p.m. and fired up at 5 a.m. excepting when the outside temperature is below 25 F when the daytime operation continues for 24 hr.

Minneapolis: 24-hour operation is in effect but pressure is reduced and heating controls set back from 11 p.m. to 5 a.m. In summer the plant is inoperative, hot water being supplied by automatic gas-fired boilers located in hot water generating stations scattered throughout the project.

Omaha: In winter the boilers are banked between 11:30 p.m. and 5 a.m. In summer, only one boiler is used and it is banked between 8 p.m. and 7 a.m.

Cincinnati: 24-hour operation is in effect both in winter and summer. In winter the temperature controls are set back between 11 p.m. and 5 a.m. In summer only one boiler is used.

1469

No. 1469

FRICTION LOSSES IN ROUND ALUMINUM DUCTS

By F. W. HUTCHINSON*, BERKELEY, CALIF.

BECAUSE OF THE uncertainties in the then existent duct friction charts, the A.S.H.V.E. Technical Advisory Committee on Air Distribution and Air Friction initiated studies in 1945 leading to development of the two friction charts that now appear in the HEATING VENTILATING AIR CONDITIONING GUIDE.[1] These two charts were developed by Wright[2] after reviewing the literature and selecting—from test work by others—that roughness value, 0.0005 ft, which seemed most applicable to round, galvanized, sheet metal duct with approximately 40 joints per 100 ft. For duct materials having greater or less roughness than galvanized iron, Wright stated that *The best solution to the problem would be to have several different charts for the materials most frequently used.* Wright also expressed uncertainty with respect to the selected value of the roughness, noting that *The uncertainty lies in whether the absolute roughness, e, of the duct surface is constant for all sizes or increases with diameter.*

In a subsequent research paper[3] Madison and Elliot presented a graphical means of applying a rational correction factor to THE GUIDE charts when used with materials having a known surface roughness differing from the value selected by Wright. The Madison and Elliot contribution offered a convenient and effective method of expanding the usefulness of the basically sound Wright charts. The Madison and Elliot correction was adopted for use in THE GUIDE[4]. On this correction chart the curves for medium smooth are based on an absolute roughness of 0.00015 whereas those for very smooth are based on an absolute roughness of 0.0000015.

In practice the use of the Madison and Elliot chart as a means of correcting friction losses in THE GUIDE is sometimes difficult, since the absolute roughness of the proposed duct material may not be known and an accurate comparative

* Professor of Mechanical Engineering, University of California. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

estimate of roughness may be very difficult. The present research paper is the first of a probable series intended to provide accurate experimental data on some of the materials more commonly used in duct construction. This first paper reports limited experimental work which confirmed Wright's estimate of the roughness of galvanized iron and presents more comprehensive experimental

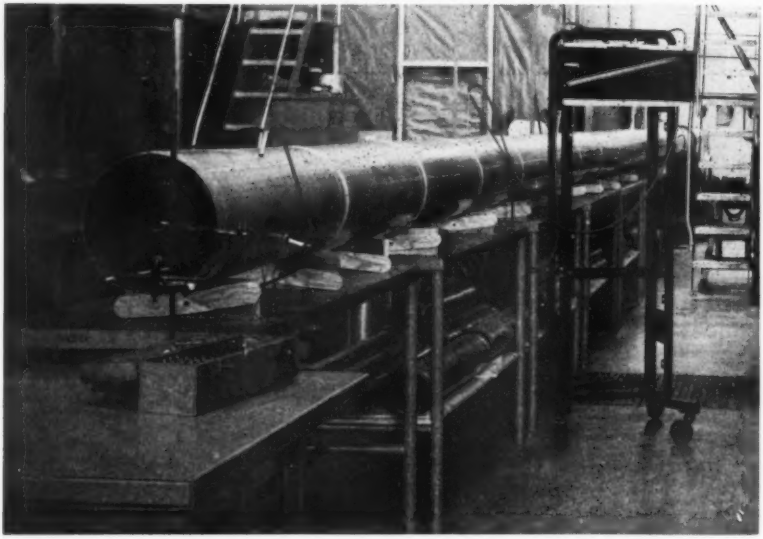


FIG. 1. GENERAL VIEW OF THE EXPERIMENTAL SET-UP WITH 12-IN. ALUMINUM DUCTS

results leading to the establishment of precise correction factors for use with THE GUIDE friction charts[1] when using aluminum as a duct material.

EXPERIMENTAL PROCEDURE

For straight round duct of uniform cross-section the problem of experimentally determining friction losses is one requiring accurate evaluation of two characteristics: (1) The average velocity (or the volume of flow) for use in the equation $Q = VA$ where Q is the flow rate in cubic feet per minute, V is the average velocity in feet per minute, and A is the cross-section area of the duct in square feet, and (2) the drop in static pressure (since for uniform cross-section the velocity pressure does not change) between points on the duct that are a known distance apart.

Fig. 1 shows a general view of the experimental set-up used in the work reported here. As a first approximation, and to assist in adjusting the variable speed of the blower, the velocity at the center of the duct was determined by means of a thermoanemometer (shown in Fig. 1 at the discharge end of the 12 in. aluminum duct). A re-check on maximum and average discharge velocity

was also obtained by using a Velometer with fittings permitting velocity evaluation a few inches in from the end cross-section of the duct.

The primary means of velocity determination was by a pitot tube traverse (both vertically and horizontally) to obtain an accurate representation of the velocity pattern across a particular cross-section. The pitot tube is shown in Fig. 1 inserted in the test section approximately 10 ft upstream from discharge. Total pressure and static pressure connections are shown going to an inclined differential pressure manometer which read velocity pressure. The manometer was graduated to hundredths of an inch and could be read with an accuracy of approximately 0.003 in. of water. Based on the pitot traverses the average velocity could be evaluated from the equation,

$$VA = V'dr^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

V = average velocity expressed in feet per minute.
A = cross-section area in square feet.
r_o = radius of the duct, in feet.
dr = differential radius.
V' = point velocity, at a given pitot position, obtained from the Equation 2.

$$V' = 4005 \sqrt{p_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

V' = point velocity, feet per minute.
p_v = pitot-measured velocity pressure, inches of water. All readings were corrected to standard air.

Static pressure taps were circumferentially located at a cross-section 18 in. upstream from discharge and at another cross-section approximately 38 ft upstream of the first (the exact distance between taps varied slightly with different sizes of duct). Both sets of taps are shown in Fig. 1, the two being connected (not shown) by tubing to a differential inclined manometer which read loss of static pressure between the test cross-sections. This manometer also could be read with an accuracy of approximately 0.003 in. of water. At various times during the test period the relative position of the static pressure taps with respect to entrance and discharge was altered and the resultant pressure loss per 100 ft of duct compared with the value obtained when the taps were in the original position. This was done in order to assure the absence of error due to possible entrance or discharge effects. At no time, during these comparative tests, was any difference in static loss (per 100 ft) observed which was great enough to be attributable to other than the limit of experimental accuracy.

As shown in Fig. 1 the 12-in. test duct is in place. Beneath the tables in the same figure can be seen the 6-in. galvanized, 6-in. aluminum, and 3-in. aluminum ducts which were installed for later tests.

Fig. 2 shows the fan with variable-speed drive which was used for tests at low-to-moderate velocity. For the high velocity tests the fan shown was replaced with a high-speed blower direct-connected to a motor and operating therefore at fixed speed. Shown in Fig. 2 is a typical transition piece connecting the fan discharge to a 6-ft section of duct, of test diameter, in which *egg crate* straightening lattices were installed. On tests with 3-in., 6-in., and 12-in. duct the

straightening section was, in each case, 6 ft long and contained two separated 2½-ft lattice straighteners.

Following the standard practice with aluminum duct work the sections were 3 ft in length, thus giving approximately 33 joints per 100 ft of length. Joints (see Fig. 1) were of standard slip type with the crimped end of one section

FIG. 2. FAN WITH VARIABLE SPEED DRIVE FOR SUPPLYING AIR FOR TESTS

being slipped into the uncrimped end of the adjacent downstream section; all joints were then externally taped as shown in Fig. 1.

ANALYSIS OF EXPERIMENTAL RESULTS

Since THE GUIDE friction loss charts are based on variable friction factors as determined from the Moody[5] chart, Fig. 3, the procedure in the present work was to use the experimentally determined static pressure loss and velocity (for a given size duct) to obtain from the Moody chart the corresponding value of the relative roughness, e/D; the absolute roughness, e, could then be readily calculated in terms of known diameter. Table 1 presents a summary of results and refers to Appendix A for typical sample calculations showing the method followed in evaluation of e/D and of the absolute roughness e.

The significance of results shown in Table 1 becomes more readily apparent when the e/D values from the Table are shown graphically—with respect to typical materials—on the Moody[5] representation of Fig. 4. The three points marked with x each indicate one experimental determination of e/D for 6-in.

galvanized duct. One point (from test 9 at 1310 fpm velocity) exactly agrees with Wright's[2] assumption of 0.0005 as the roughness for galvanized iron whereas the other two points bracket the first, but do not indicate appreciable departure. Based on this limited experimental work it is suggested that Wright's selection of 0.0005 for galvanized iron duct is in accord with the roughness of such ducts of average construction; in effect, this limited work confirms the accuracy of the A.S.H.V.E. friction chart when used—as was intended—with galvanized iron.

The experimental points on Fig. 4 for the tests on aluminum duct indicate a substantially smaller value of the absolute roughness. For 3-in. duct one point

TABLE 1—EXPERIMENTAL RESULTS

TYPE OF DUCT	TEST NO.	DUCT DIAMETER INCHES	VELOCITY[a] Fpm	HEAD LOSS[b] IN. WATER	e/D[c]	e[d]
Aluminum	1	3	1505 fpm	1.34	0.0006	0.00015
	2	3	4710	11.00	0.0005	0.00012
	3	6	1325	0.495	0.0003	0.00015
	4	6	2002	0.99	0.0004	0.0002
	5	6	5940	7.07	0.0002	0.0001
	6	12	1975	0.40	0.0001	0.0001
	7	12	2310	0.533	0.000186	0.000186
	8	12	2740	0.78	0.0002	0.0002
Galvanized Iron	9	6	1310	0.493	0.001	0.0005
	10	6	1932	1.014	0.0009	0.00045
	11	6	5850	9.24	0.0013	0.00065

[a] The recorded velocity for each test is the average value as determined by a pitot tube traverse of the duct.
[b] The head loss is expressed in inches of water per 100 ft length of straight duct; actual test sections average 37 ft between static pressure taps.
[c] Sample calculation (for test 9) given in Appendix A.
[d] Sample calculation (for test 9) given in Appendix A.

exactly corresponds to the value of e/D for commercial steel or wrought iron pipe (with an absolute roughness of 0.00015) and the second point—at much higher velocity—indicates a slightly lower (0.00012) value. For 6-in. duct one point again exactly corresponds to the e/D for commercial steel and wrought iron pipe whereas the other two points bracket the first. For 12-in. duct the three experimental points bracket the e/D for commercial steel pipe, but do not differ greatly from it.

DISCUSSION OF RESULTS

From Table 1 and the plotting of points on Fig. 4 it seems evident that the absolute roughness of aluminum duct is of the order of 1/3 that of galvanized iron (absolute roughness of approximately 0.00015 as compared with 0.0005). Thus in designing aluminum ductwork proper correction factors should be applied to friction losses as determined from the charts in THE GUIDE.

As a first approximation THE GUIDE correction curve[4] for medium smooth material could be used. Slightly more conservatively, however, and to permit

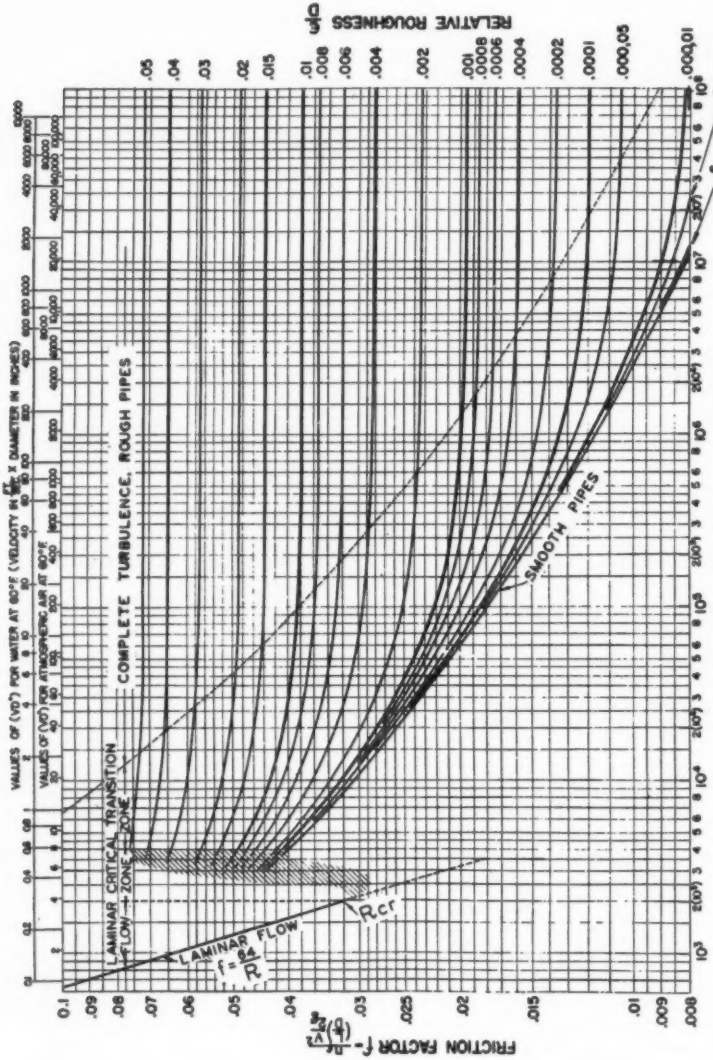

VALUES OF (VD') FOR WATER AT 60°F (VELOCITY IN FT/SEC, D = DIAMETER IN INCHES)
VALUES OF (VD') FOR ATMOSPHERIC AIR AT 60°F
COMPLETE TURBULENCE, ROUGH PIPES
LAMINAR CRITICAL TRANSITION ZONE
LAMINAR FLOW
f = 64/R
Rcr
SMOOTH PIPES
RELATIVE ROUGHNESS ε/D
FRICTION FACTOR f = D/(V²/2g)
REYNOLDS NUMBER R = VD/ν (V IN FT/SEC, D IN FT, ν IN FT²/SEC)

FIG. 3. RELATION BETWEEN FRICTION FACTOR AND REYNOLDS' NUMBER
(Reproduced from *A.S.M.E. Transactions*, November 1944)

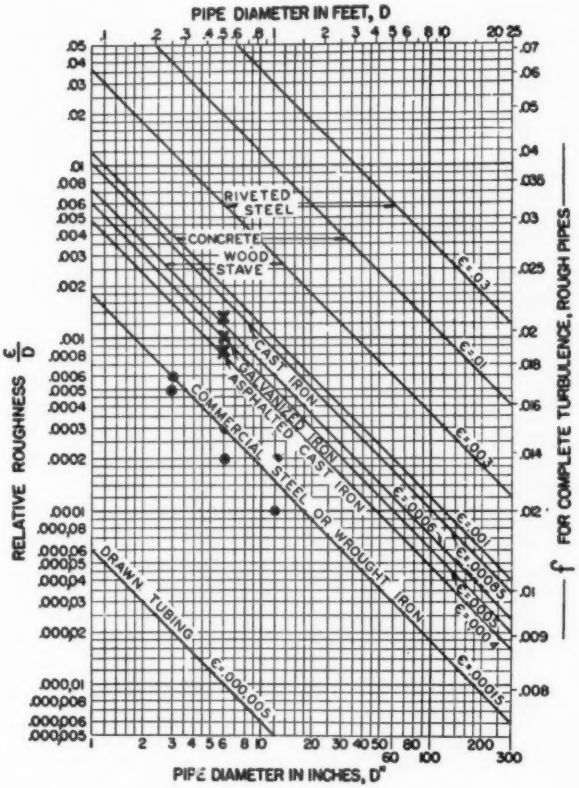


FIG. 4. THE MOODY REPRESENTATION COMPARING RELATIVE ROUGHNESS *(e/D)* FOR SEVERAL MATERIALS

• = Values for aluminum determined by experiments
x = Values for galvanized iron determined by experiments

(Reproduced from *A.S.M.E. Transactions*, November 1944)

greater accuracy over a wider range of diameters specific recommended correction factors have been computed for aluminum duct (based on the experimental results here reported) and are presented graphically (in form identical with that used in THE GUIDE's method of Madison and Elliot[4]).

In developing the correction factors it was noted (Table 1) that for duct sizes of 3-in. and 6-in. a design value of absolute roughness for aluminum of 0.00015 would appear acceptable. For duct sizes greater than 12-in. there seems to be a possibility that the absolute roughness exceeds 0.00015, but it is thought conservative to use 0.0002 as a design value. As noted by Wright[2] inadequate

data are available to permit determination concerning the possible increase of the absolute roughness with diameter, but certain theoretical considerations suggest that a slight increase might be expected. Thus with round duct and slip joints the *roughness* responsible for loss of head is of two kinds: (1) Roughness of the material of which the duct is constructed and (2) roughness due to discontinuities of the internal surface at each joint. Fig. 5 shows clearly the

FIG. 5. END VIEW OF DUCTS SHOWING MARKED DISCONTINUITIES AT JOINTS

(Prepared and assembled under ordinary shop conditions)

marked discontinuities that occur at joints of duct prepared and assembled under ordinary shop conditions†.

The foregoing two types of roughness superimpose and combine to establish the experimentally determined *absolute roughness*. The roughness of the material would be expected to remain constant, but the joint roughness might well vary as a function of diameter due to the fact that—for fixed 3 ft length of sections—the number of diameters between joints decreases rapidly as the duct diameter increases. In connection with another type of problem this concept of composite roughness has been investigated by Einstein and Banks[6]. Thus to assure con-

† In order to insure that the duct tested on the present research project would conform to average practice the duct sections were fabricated by a commercial sheet metal establishment and the method of assembling and taping sections corresponded to actual practice. Thus irregularities and discontinuities at the joints were substantially greater than would normally have occurred in laboratory work.

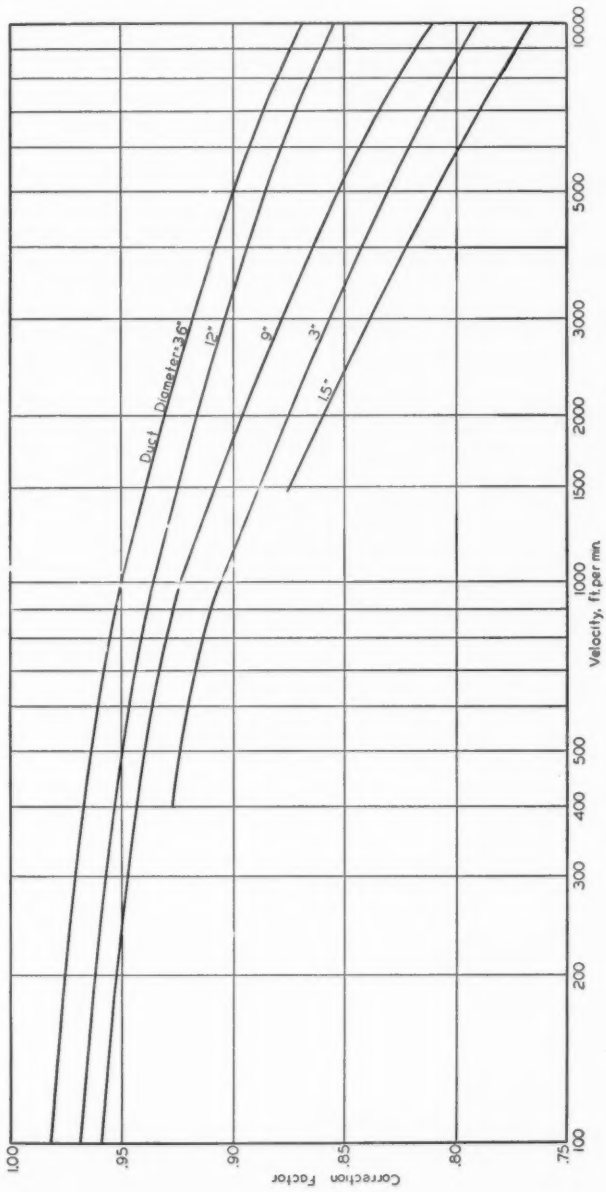


FIG. 6. CORRECTION FACTORS FOR ALUMINUM DUCT

servative results the correction factors for duct diameters greater than 10-in. have been based on an absolute roughness of 0.0002 as against the value of 0.00015 for ducts 10-in. in diameter or smaller.

Fig. 6 presents in graphical form the recommended correction factors, for aluminum ductwork, to be used with THE GUIDE friction charts[1]. The calculations, on which Fig. 6 is based, were made from experimental results tabulated in Table 1; a typical sample calculation of a correction factor is given in detail in Appendix B.

Of particular interest is the fact that friction losses in aluminum duct work are from 15 to 25 percent less than in galvanized ducts for applications where the diameter is small and the velocity is high. Since there seems to be growing adoption of high velocity design it seems particularly important that accurate correction factors be made available in these regions where uncorrected use of THE GUIDE friction charts would lead to overly conservative results.

CONCLUSIONS

The A.S.H.V.E. duct friction charts[1] were designed for galvanized iron and based on a value of the absolute roughness of 0.0005 ft. For use with ducts made of materials other than galvanized iron THE GUIDE charts require correction and this is usually accomplished by application of factors taken from THE GUIDE correction curves[4]. Unfortunately accurate experimental data have not been available on the absolute roughness of materials used in duct work other than galvanized iron; hence the practical designer has had difficulty in selecting the proper correction coefficient. The present research report provides limited experimental data which further corroborate the accuracy of THE GUIDE friction charts. It also presents detailed experimental data leading to the development of recommended correction coefficients (Fig. 6) for use with THE GUIDE charts when using aluminum as the duct material.

The test results indicate that the existing GUIDE correction factors[4] for a *medium smooth* material may be used for aluminum with reasonable accuracy, but the values given in Fig. 6—though slightly more conservative at larger diameters—are more accurate because they include a larger range of diameters, hence requiring less interpolation.

APPENDIX A

EVALUATION OF ROUGHNESS

The HEATING VENTILATING AIR CONDITIONING GUIDE 1952 (p. 672) states that the pressure loss in circular ducts is given by the equation:

$$h_f = f\,(l/D)\,(v^2/2g) \quad \text{(A-1)}$$

where

h_f = head loss due to friction, in feet of the fluid flowing.
l = length of conduit, feet.
D = inside diameter of conduit, feet.
v = mean fluid velocity, feet per second.
g = acceleration due to gravity, 32.17 feet per (second) (second).
f = a non-dimensional friction coefficient, which for ventilation work depends on the Reynolds number and the relative roughness of the conduit.

Taking the specific weight of water as 62.3 lb per cu ft and the density of air as 0.075 lb per cu ft a relationship can be established between the head loss in feet of air flowing and the equivalent head loss, H, expressed in inches of water:

$$H = (0.075 \times 12)/(62.3)\, h_t$$

or,

$$h_t = (62.3 \times H)/(0.075 \times 12) = f(l/D)\,(v^2/2g).$$

For a duct length of 100 ft, with the velocity expressed in the more convenient units of feet per minute, V, and the diameter in the more convenient units of inches, D inches, Equation A-1 permits evaluation of the friction factor, f, as

$$f = (62.3 \times 64.4 \times H \times D'' \times 3600) \div (0.075 \times 12 \times 100 \times 12 \times V^2)$$
$$= 13{,}370(HD''/V^2) \quad \text{(A-2)}$$

where

f = friction factor.
H = head loss per 100 feet expressed in inches of water.
D'' = duct diameter in inches.
V = velocity in feet per minute.

Sample Calculation: From Table 1, test 9 on 6 in. galvanized pipe at 1310 fpm gave a static pressure loss of 0.493 in. per 100 ft of duct. The friction factor is then,

$$f = 13{,}370(0.493 \times 6/1310^2) = 0.0230 \text{ and the } VD'' \text{ product is } (1310/60)\,(6) = 131.$$

Then from Fig. 3 with $f = 0.023$ and $VD'' = 131$ the value of e/D is read as 0.001 and the absolute roughness, e, is then $(6/12)\,(0.001) = 0.0005$. This value is in agreement with the assumed absolute roughness selected by Wright[2] and used in constructing THE GUIDE friction charts for flow through galvanized iron ducts.

APPENDIX B

EVALUATION OF CORRECTION FACTOR

In THE GUIDE the roughness used in construction is 0.0005. Thus for a 3-in. diameter duct the value of $e/D = 0.005/0.25 = 0.0020$. At a selected velocity of 10,000 fpm the VD'' product is $(10{,}000/60)\,(3) = 500$, so entering Fig. 3 at a VD'' of 500 and an e/D of 0.0020 one reads the corresponding friction factor as 0.024.

With aluminum duct work at 3-in. diameter the roughness should be taken as 0.00015 so the corresponding $e/D = 0.00015/(3/12) = 0.00060$ and the VD'' product, at 10,000 fpm velocity, remains 500; therefore, from Fig. 3 the corresponding friction factor is read as 0.0190.

The correction factor to be applied to friction losses read from the chart in THE GUIDE is then the ratio of the friction factor for aluminum to the friction factor used in THE GUIDE: thus for 3 in. aluminum duct with 10,000 fpm velocity, the correction factor is $0.0190/0.024 = 0.792$ which indicates that the actual friction loss for the aluminum duct is more than 20 percent less than that through the galvanized duct for which the A.S.H.V.E. chart is constructed.

REFERENCES

1. HEATING VENTILATING AIR CONDITIONING GUIDE 1952, Chapter 31—Air Duct Design, pp. 670-671, Figs. 1 and 2 (published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.).

2. A.S.H.V.E. RESEARCH REPORT No. 1280—A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 303).

3. Friction Factors for Gases Including Correction for Temperature, Viscosity and Pipe Roughness, by R. D. Madison and W. R. Elliot (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning,* October 1946).

4. HEATING VENTILATING AIR CONDITIONING GUIDE 1952, Chapter 31—Air Duct Design, p. 673, Fig. 3 (published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.).

5. Friction Factors for Pipe Flow, by L. F. Moody (*A.S.M.E. Transactions,* Vol. 66, 1944, p. 671.)

6. Fluid Resistance of Composite Roughness, by H. A. Einstein and R. B. Banks (*Transactions* of the *American Geophysical Union,* Vol. 31, August 1950, p. 603).

1470

No. 1470

PRESSURE LOSSES IN 4-INCH DIAMETER GALVANIZED METAL DUCT AND FITTINGS

By H. G. Conn*, W. G. Colborne**, and W. G. Brown†,
Kingston, Ont., Canada

THE TREND toward the use of small pipes in warm-air heating and air conditioning has increased the need of accurate values of pressure loss in these pipes and their fittings, especially in the fittings and combinations of fittings. This initial investigation deals with the pressure losses in 4-in. pipe and fittings.

Apparatus

A major difficulty was the accurate measurement of the air flow in the range desired. The range of air flow was from 40 to 80 cfm of standard air, having a density of 0.075 lb per cu ft. The method of air measurement adopted was that given in the *A.S.M.E.* Power Test Code for Displacement Compressors[1], the ¾-in. screw type standard nozzle being used. The air was supplied by a two stage air compressor with a capacity of 290 cfm. Fig. 1 shows the arrangement of the apparatus. In order to maintain a constant pressure drop across the nozzle and hence a constant air flow during any given set of readings, it was necessary to keep the pressure in the receiver tank at approximately 95 psig. At this pressure, the air compressor unloaders which operate at 100 psig, remained inoperative and the air compressor therefore ran continuously at full load. By adjusting the pressure reducing valve and the take-off valve, the air

* Head, Department of Mechanical Engineering, Queen's University.

** Researcher, Queen's University.

† Assistant Researcher, Queen's University.

[1] Exponent numerals refer to Bibliography.

Presented at the 59th Annual Meeting of The American Society of Heating and Ventilating Engineers, Chicago, January 1953.

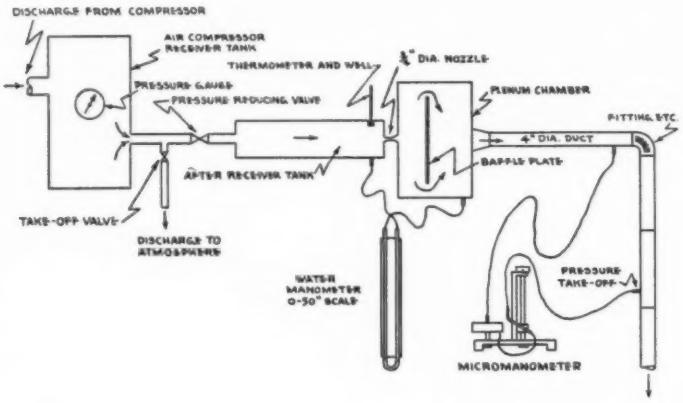


FIG. 1. SCHEMATIC DRAWING SHOWING GENERAL ARRANGEMENT OF APPARATUS USED IN TESTING 4-IN. DIAMETER AIR DUCT AND FITTINGS

flow through the nozzle could be varied and the pressure in the receiver tank maintained at 95 psig.

The pressure drop across the standard measuring nozzle was indicated on a 50-in. water manometer calibrated in tenths of an inch.

The plenum chamber and baffle were installed to reduce the high velocity of the jet from the nozzle and to obtain uniform flow into the 4-in. pipe. The fitting or fittings to be tested were then installed and the pressure loss through them measured on a micromanometer as shown in Fig. 1.

AIR FLOW CALCULATIONS

The air flow in each test was calculated by the simplified formula specified in the *A.S.M.E.* Power Test Code for Displacement Compressors[1]:

$$w = 6.870 \times C D_2^2 \sqrt{\frac{P_b \Delta P}{T_2}} \quad \ldots\ldots\ldots\ldots (1)$$

Substituting for D_2 and converting to cfm of standard air Equation 1 reduces to

$$(cfm)_s = 51.5 \times C \sqrt{\frac{P_b \Delta P}{T_2}} \quad \ldots\ldots\ldots\ldots (2)$$

In order to reduce the number of calculations made in these tests, values of $P_b \Delta P / T_2$ ranging from 0.5 to 2.6 were plotted against cfm of standard air for various values of C ranging from 0.976 to 0.984.

TEST PROCEDURE

The initial tests were run to determine the pressure loss in straight 4-in. pipe. The tests on the fittings were then run locating the upstream pressure taps 7

NOMENCLATURE

w = rate of air flow, pounds per minute.
C = coefficient of discharge, non-dimensional.
D_2 = throat diameter of nozzle, inches.
P_b = barometric pressure, inches of mercury.
ΔP = differential pressure across nozzle, (inches water gage).
T_2 = absolute temperature upstream from the nozzle, (Fahrenheit, absolute).
H = pressure loss, inches of water.
h = pressure loss, feet of fluid flowing.
V_s = velocity, feet per second.
V = velocity, feet per minute.
f = friction coefficient, non-dimensional.
L = length of pipe, feet.
D = diameter of pipe, feet.
g = acceleration due to gravity, feet per second per second.
ρ = air density, pounds per cubic foot.
ν = kinematic viscosity of air, square feet per second.
d = diameter of pipe, inches.
E = friction loss, equivalent length in feet of pipe.
R/D = ratio of center line radius to the diameter of an elbow.
K and a = dimensionless constants.
R_e = Reynold's number, dimensionless.
cfm = air flow, cubic feet per minute.
P_v = velocity pressure, inches water gage.
$(\text{cfm})_s$ = air flow, standard air, cubic feet per minute.

diameters ahead of the fitting and the downstream pressure taps, 21 diameters after the fitting, in order to eliminate any interference caused by the fitting. The pressure loss of the straight pipe was subtracted from the total pressure loss reading, giving the pressure loss of the fitting itself. While it is realized that this is not necessarily the loss *in* the elbow, it does represent the loss *due to* the elbow.

A simplified correction formula to correct for the variation in density was derived as follows:

The Fanning equation gives

$$h = \frac{fL}{D} \cdot \frac{V_s^2}{2g} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

which for a particular case reduces to

$$H = K_1 f V^2 \rho \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

where H is now expressed in inches of water and V is in feet per minute. For air under standard conditions

$$H_s = K_1 f_s V_s^2 \rho_s \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

therefore

$$\frac{H}{H_s} = \frac{f}{f_s} \times \frac{\rho}{\rho_s} \times \left(\frac{V}{V_s}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

For the same cfm and hence the same velocity

$$\frac{H}{H_s}=\frac{f}{f_s}\times\frac{\rho}{\rho_s} \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

If the assumption be made that the curve of f against R_e is a straight line on log paper in the small range being discussed, then Equation 7 can be expressed

$$\frac{H}{H_s}=\left(\frac{\nu}{\nu_s}\right)^a\times\frac{\rho}{\rho_s} \quad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

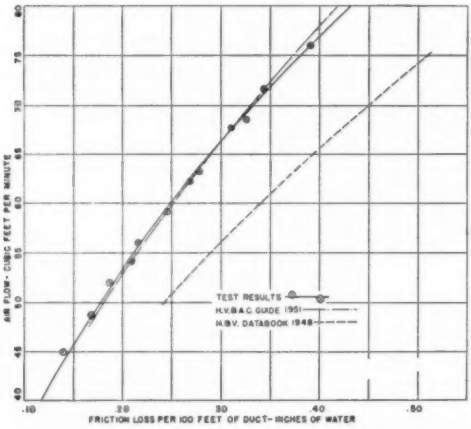

AIR FLOW - CUBIC FEET PER MINUTE
FRICTION LOSS PER 100 FEET OF DUCT - INCHES OF WATER
TEST RESULTS
H.V.B.A.C. GUIDE 1951
H.B.V. DATABOOK 1948

FIG. 2. FRICTION LOSSES IN 4-IN. DIAMETER GALVANIZED IRON AIR DUCT HAVING 40 JOINTS PER 100 FT

By measurement the relationship between f and R_e was determined for the straight 4-in. pipe, and this relationship showed that $a=+0.12$

therefore

$$\frac{H}{H_s}=\left(\frac{\nu}{\nu_s}\right)^{+0.12}\times\frac{\rho}{\rho_s} \quad \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

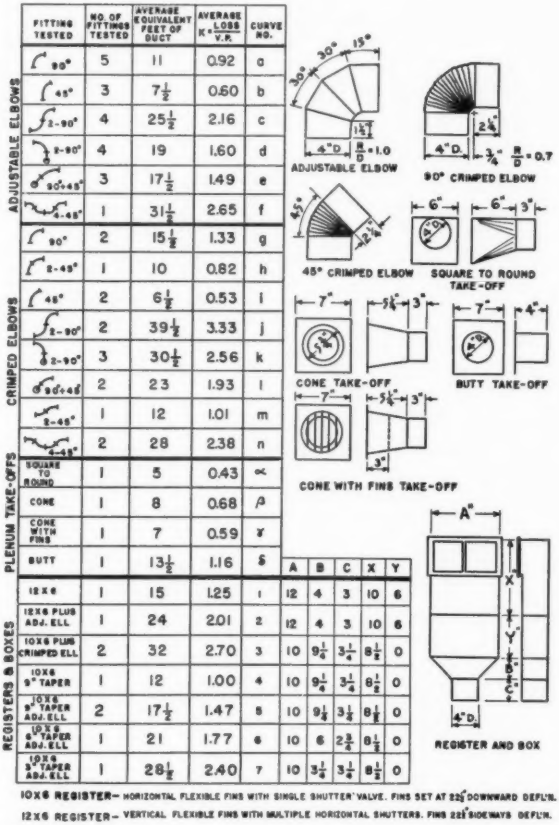
During the experiments the maximum variation in temperature was ± 5 deg from 73 F. For this variation the kinematic viscosity term could be neglected and H_s then becomes

$$H_s=H\frac{\rho_s}{\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots (10)$$

which is based on a constant cfm.

RESULTS

The results obtained for the pressure loss in 4-in. diameter pipe correspond to the values given in THE GUIDE 1951[2] (see Fig. 2). From the paper by



	FITTING TESTED	NO. OF FITTINGS TESTED	AVERAGE EQUIVALENT FEET OF DUCT	AVERAGE LOSS K = LOSS / V.P.	CURVE NO.
ADJUSTABLE ELBOWS	90°	5	11	0.92	a
	45°	3	7½	0.60	b
	2-90°	4	25½	2.16	c
	2-90°	4	19	1.60	d
	90°+45°	3	17½	1.49	e
	4-45°	1	31½	2.65	f
CRIMPED ELBOWS	90°	2	15½	1.33	g
	2-45°	1	10	0.82	h
	45°	2	6½	0.53	i
	2-90°	2	39½	3.33	j
	2-90°	3	30½	2.56	k
	90°+45°	2	23	1.93	l
	2-45°	1	12	1.01	m
	4-45°	2	28	2.38	n
PLENUM TAKE-OFFS	SQUARE TO ROUND	1	5	0.43	∝
	CONE	1	8	0.68	β
	CONE WITH FINS	1	7	0.59	ɣ
	BUTT	1	13½	1.16	δ

	REGISTERS & BOXES					A	B	C	X	Y
	12X6	1	15	1.25	1	12	4	3	10	6
	12X6 PLUS ADJ. ELL	1	24	2.01	2	12	4	3	10	6
	10X6 PLUS CRIMPED ELL	2	32	2.70	3	10	9¼	3¼	8½	0
	10X6 9° TAPER	1	12	1.00	4	10	9¼	3¼	8½	0
	10X6 9° TAPER ADJ. ELL	2	17½	1.47	5	10	9¼	3¼	8½	0
	10X6 6° TAPER ADJ. ELL	1	21	1.77	6	10	6	2¾	8½	0
	10X6 3° TAPER ADJ. ELL	1	28½	2.40	7	10	3¼	3¼	8½	0

10X6 REGISTER— HORIZONTAL FLEXIBLE FINS WITH SINGLE SHUTTER VALVE. FINS SET AT 22½° DOWNWARD DEFL'N.

12X6 REGISTER— VERTICAL FLEXIBLE FINS WITH MULTIPLE HORIZONTAL SHUTTERS. FINS 22½° SIDEWAYS DEFL'N.

FIG. 3. FRICTION LOSS DATA FOR 4-IN. DIAMETER GALVANIZED IRON AIR DUCT FITTINGS

D. K. Wright and the discussion by R. D. Madison[3] the equation developed to give the pressure drop was

$$\Delta P = 2.74 \frac{L}{d^{1.22}} \left(\frac{V}{1000} \right)^{1.9} \quad \ldots\ldots\ldots\ldots\ldots (11)$$

For the particular case of 4-in. diameter duct, with length 100 ft this reduces to

$$\Delta P = \frac{10.08}{10^5} V^{1.9} \quad \ldots\ldots\ldots\ldots\ldots (12)$$

The equation of the curve developed from the test results in Fig. 2 is

$$H = \frac{11.3}{10^5} V^{1.88} \quad \ldots\ldots\ldots\ldots\ldots (13)$$

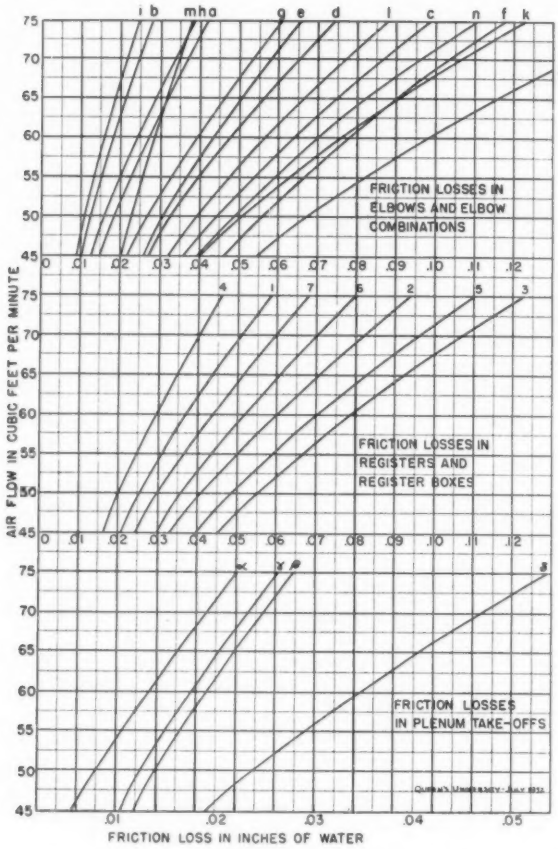

i b mha g e d l c n f k
j
FRICTION LOSSES IN ELBOWS AND ELBOW COMBINATIONS
4 1 7 6 2 5 3
FRICTION LOSSES IN REGISTERS AND REGISTER BOXES
FRICTION LOSSES IN PLENUM TAKE-OFFS
AIR FLOW IN CUBIC FEET PER MINUTE
FRICTION LOSS IN INCHES OF WATER

FIG. 4. FRICTION LOSS CURVES FOR 4-IN. DIAMETER AIR DUCT FITTINGS

The small difference in these equations could be accounted for in the interpretation of the points from the test results.

The description and arrangement of the fittings tested are shown in Fig. 3. The resistance of the fittings is expressed in terms of the number of velocity pressures lost and also as equivalent feet of straight pipe. These values are also given in Fig. 3. The curves showing the exact losses at various air flow rates are given in Fig. 4.

To facilitate the reading of the curves the actual points showing the results of individual tests were not shown in Fig. 4. Fig. 5 was plotted showing the actual

points and illustrates the *scatter* that may be expected when testing different elbows of the same size and shape.

The 4-piece adjustable elbows as received had flakes of galvanizing on the inside at the places where the joints were located. These flakes still adhered

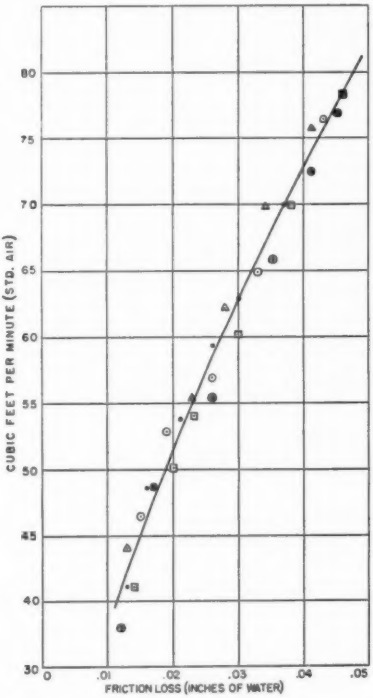


FIG. 5. MEAN VALUES OF FRICTION LOSSES IN 5 DIFFERENT 4-PIECE ADJUSTABLE 90-DEG ELBOWS

to the metal but were protruding slightly, increasing the surface roughness. The results shown in Fig. 5 were obtained by using the elbows as they were received.

The flakes of galvanizing were then removed from the elbows with emery cloth, and the tests were re-run. These test results showed a decrease of 20 percent in the pressure loss value.

An investigation into the effect of rotating the second elbow of a pair of 90 deg elbows was carried out, for both crimped and 4-piece adjustable elbows.

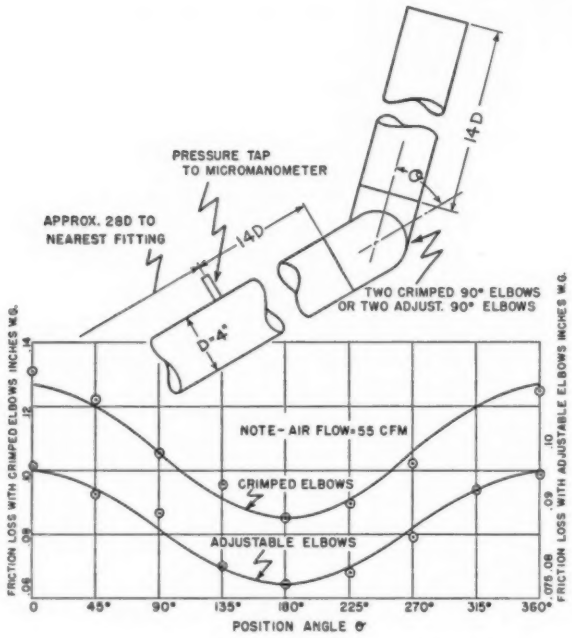


FIG. 6. SINUSOIDAL LOSS VARIATIONS FOR 2 ELBOW COMBINATIONS

The results (see Fig. 6) approximated a sinusoidal variation of pressure loss with angle of rotation.

The equation for the pressure loss in the elbow combination has the form

$$H = K \cos \theta + a \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (14)$$

where

K and a are constants.

Knowing the pressure loss with the elbows arranged at any two angles, the constants in Equation 14 can be calculated. The pressure loss for two elbows in combination at any angle can then be found.

The equations calculated from the results obtained and expressed in equivalent feet of pipe are as follows:

For two 90 deg crimped elbows

$$E = 9 \cos \theta + 30.5 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (15)$$

For two 90 deg 4 piece adjustable elbows

$$E = 6.5 \cos \theta + 19.0 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (16)$$

Further work is required using elbows of various diameters and R/D ratios in order to verify this relationship under other conditions. It is felt, however, that a reasonably accurate pressure loss can be found for two elbows at various angles using an equation of the form of Equation 14.

In the tests run on the registers and register boxes the pressure loss figures include the velocity pressure at the register face. These pressure loss values are therefore the sum of the static pressure and the velocity pressure at the point from which the loss is measured. Fig. 7 illustrates the location of the pressure taps in the register tests.

The next investigation was carried out on runs of 4-in. pipe containing several fittings such as elbows and elbow combinations. The purpose was to study the

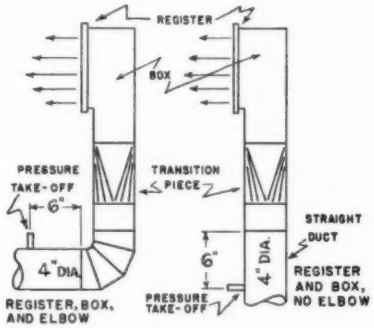


FIG. 7. PRESSURE TAKE-OFF LOCATIONS FOR FRICTION LOSS TESTS ON REGISTERS

difference, if any, between the pressure loss calculated from the results obtained for each separate fitting and the measured pressure loss on these composite runs. Hereinafter, the pressure loss for a composite run, when calculated as the sum of the losses for the individual fitting and pipe as previously found, will be referred to as the predicted loss.

For 25 composite runs tested it was noted that on the average the measured pressure loss for the run was 10 percent lower than the predicted pressure loss. The maximum variations occurred in runs with less than 15 diameters of straight pipe between fittings. One of these more critical composite runs had a measured pressure loss 30 percent lower than the predicted pressure loss, while another one had the measured loss 10 percent higher than the predicted loss. In the various studies made, no consistent method of predicting the probable variation could be found.

Total pressure profiles were run at various cross-sections along the pipe downstream from elbow combinations in an effort to relate the uneven velocity in the pipe to the variations of the measured from the predicted losses. Five such total pressure profiles are shown in Fig. 8 and the arrangement of the

duct is shown in Fig. 9. The numbers on the curves refer to the numbers of the sections as shown in Fig. 9.

Total pressure profiles were run only in the horizontal plane in order to obtain an indication of the velocity distribution rather than a true average of the total pressure over the area of the pipe.

The measured pressure loss between stations 1 and 5, Fig. 9, was 18.8 percent lower than the predicted pressure loss. By the use of a planimeter the difference

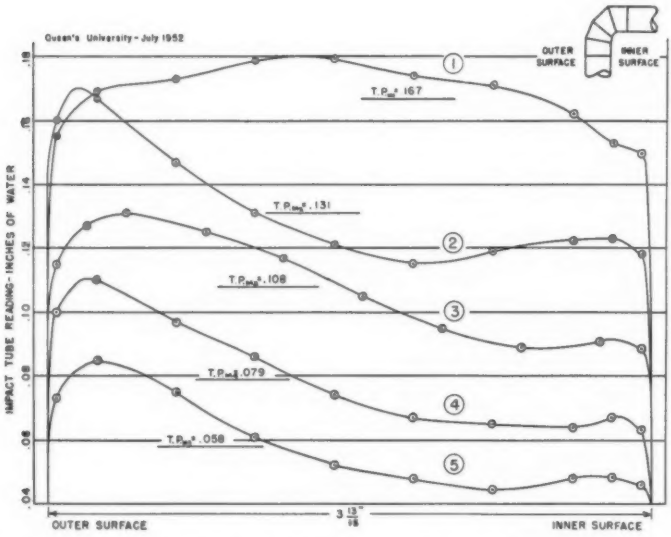


FIG. 8. TOTAL HEAD PROFILES AT 5 POSITIONS IN THE SAME DUCT SYSTEM AT 58 CFM AIR FLOW

of the areas under curves 1 and 5 was found and this difference was 19.3 percent lower than the predicted loss which checks closely with the 18.8 percent difference found by measuring the static pressure difference.

A study of the arrangement in Fig. 9, and others, failed to produce a satisfactory explanation of the difference between the measured and predicted losses.

It seems apparent that a future project might be the study of the effect of one fitting on the next succeeding fitting, and others, downstream.

CONCLUSIONS

1. The pressure loss in 4-in. diameter pipe as determined in these tests agrees with the values given in THE GUIDE[2] in the range tested.

2. A portion of the pressure loss attributed to an elbow occurs in the pipe following the elbow. The pressure loss of an elbow, therefore, depends partially on what fitting or pipe follows that elbow. The pressure loss of an elbow also depends on the velocity

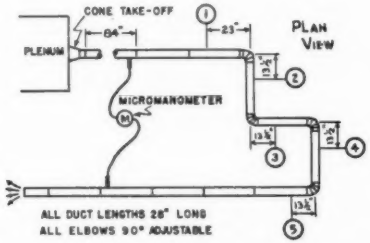


FIG. 9. ARRANGEMENT FOR IMPACT TUBE PROFILE TEST

distribution in the stream entering the elbow and therefore depends on the fittings immediately before that elbow.

The equivalent length of one of the 4-piece adjustable elbows varied from 5½ ft when used in combination with a register box, to 14½ ft when used in combination with another elbow. The elbow when tested singly had a pressure loss equivalent to 11 ft of pipe.

3. A recent proposal[4] of stating the pressure loss in an elbow in terms of the surface roughness and the R/D ratio is unsatisfactory in cases where the elbow has joints or crimps. In these cases it is very difficult to estimate the surface roughness.

4. The pressure loss expressed in equivalent feet of straight pipe is the most satisfactory method of evaluating the loss for a composite run. The actual pressure loss

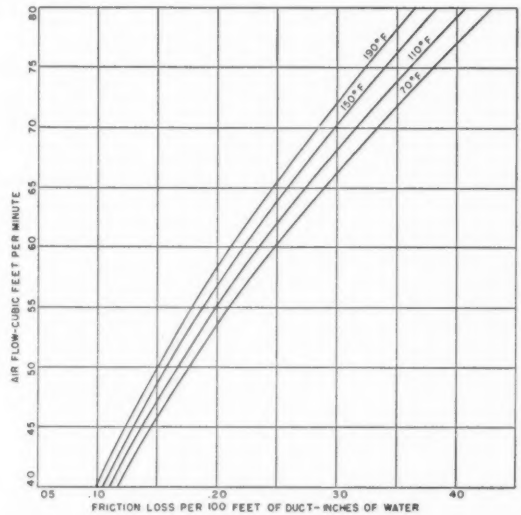


FIG. 10. FRICTION LOSSES FOR AIR FLOW IN STRAIGHT DUCT AT VARIOUS TEMPERATURES AND 29.92 IN. HG BAROMETRIC PRESSURE

expressed in inches of water can then be found for various temperatures as shown in Fig. 10.

5. When two or more fittings are immediately joined the pressure loss should be evaluated, treating the combinations as one single fitting.

BIBLIOGRAPHY

1. *A.S.M.E.* Power Test Code for Displacement Compressors, Vacuum Pumps and Blowers, May 1939.

2. HEATING VENTILATING AIR CONDITIONING GUIDE 1951 (published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 62 Worth St., New York 13, N. Y.).

3. A.S.H.V.E. RESEARCH REPORT No. 1280—A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 315).

4. Pressure Losses in Tubing, Pipe, and Fittings, by R. J. S. Pigott (*A.S.M.E. Transactions* 1950, p. 684).

DISCUSSION

D. W. LOCKLIN, Jackson, Mich.: It appears that the losses in Fig. 6 include components other than those due to the elbows alone. I would like to invite the authors to comment on the losses included in those data.

I believe the authors would be interested in reviewing the work of Dr. J. R. Weske in his *N.A.C.A.* report* which covers experimental data on losses in very smooth ducting. His results indicated a similar trend as reported for compound elbows in this paper; he found very little change in elbow loss when spacer lengths were added between the two elbows as shown in Fig. 6.

In composite systems, the comparison between the predicted and actual loss is very important since it gives us an indication of the applicability of our engineering calculations. I would therefore suggest that an appendix be added to summarize these comparisons for the 25 composite arrangements of fittings. This fine piece of work should certainly be recorded for reference of other researchers and designers.

AUTHOR'S CLOSURE (Professor Colborne): Referring to Fig. 6 again, as I pointed out before very briefly, the curves were plotted from the measured static pressure at the point shown as the pressure tap connection to the micromanometer. The static pressure measured at this point was, therefore, the sum of the loss in the pipe (28 diameters of pipe altogether), the loss in the elbows, and possibly some exit loss due to increased roughness at the pipe end. If the loss in the straight pipe of 28 diameters is subtracted from the static pressure the difference therefore still does not agree with the loss in two elbows as previously determined.

What we wanted to establish in this test was the form the curve would take due to rotation of the elbows. We were not trying to obtain the exact pressure loss. We did establish the shape of the curves and from that, established the equation that fitted those curves. Then using the equation, we evaluated the two constants in the equation by using previously obtained results from Fig. 3 for any two positions of the elbows.

* Pressure Loss in Ducts with Compound Elbows, by J. R. Weske (*National Advisory Committee for Aeronautics*, Advanced Restricted Report W-39, February 1943).

1471

No. 1471

ROOM AIR DISTRIBUTION RESEARCH FOR YEAR 'ROUND AIR CONDITIONING

Part I—Supply Outlets at One High Sidewall Location

By S. F. GILMAN*, H. E. STRAUB**, URBANA, ILL., A. E. HERSHEY†, EAST PITTSBURGH, PA., AND R. B. ENGDAHL††, COLUMBUS, OHIO

ENGINEERING methods have been developed for designing winter heating systems and, to a lesser degree, summer cooling systems for residences, Combining, however, of the heating and the cooling operations into a year 'round air conditioning system presents several new and unsolved problems in both duct and room air distribution[1]. In year 'round air conditioning of residences, economic reasons and operation simplicity normally require that the same supply outlets be used for both the winter heating and the summer cooling operations. Consequently, one of the problems unique of the year 'round system is that of distributing heated air in winter and cooled air in summer from the same supply outlets in such a manner that satisfactory comfort conditions are maintained at all times. Although effectiveness of the air distribution in a residence is an extremely important factor in the overall performance of the air conditioning system, engineering data for selecting proper types and locations of supply outlets utilizing both heated and cooled air are not available. The rapidly accelerating rate at which year 'round air conditioning is being installed in residences emphasizes the need for air distribution research to obtain such data.

A research project on air distribution in rooms was initiated at the University of Illinois during the latter part of 1939 and continued until April 1942, when staff members entered the Armed Forces or industries engaged in war work. The first year of the project was devoted to modifying an existing test plant and developing instruments for air distribution studies; a paper covering this

* Research Assistant Professor of Mechanical Engineering, University of Illinois. Junior member of A.S.H.V.E.

** Research Associate in Mechanical Engineering, University of Illinois. Member of A.S.H.V.E.

† Advisory Engineer, Research Laboratories, Westinghouse Electric Corp.

†† Supervisor, *Battelle Memorial Institute*. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

work has been published[2]. Since a report of research conducted between late 1940 and April 1942 has not been published, principal results obtained during that period are included in this paper.

In September 1950, the project was reactivated by the University and since June 1951 has been jointly sponsored by the *American Gas Association* and the Engineering Experiment Station of the University of Illinois. This paper also reports the work done and research conducted since September 1950.

OBJECTIVE AND SCOPE

The object of this part of the air distribution program is to obtain engineering data for selecting proper types and locations of supply outlets for satisfactory air distribution in residential rooms during all seasons of the year.

Observations were made of air temperatures, velocities, and flow patterns existing at numerous points in a test room under various winter heating and summer cooling conditions of operation. Principal variables are flow-rate (cfm); temperature and velocity of supply air; location, type, and deflection setting of supply outlets; and the location of the return intakes. Although heating load, cooling load, and room size are now being held constant, these factors may be introduced as variables later in the research program.

The research completed has involved several commonly used types and sizes of supply outlets installed at a single high sidewall location. Additional studies with supply outlets at other high sidewall locations will be conducted later in the program possibly after completing the current phase of the project which deals with floor registers of both the conventional and perimeter types.

DESCRIPTION OF APPARATUS

Since much of the equipment has been described in detail previously[2, 3], only a general description of the apparatus is given except for modifications or additions made subsequent to 1940.

A diagram of the room heating and cooling test plant is shown in Fig. 1. The test room is 13¾ ft wide and 18 ft long, and has an 8½ ft ceiling height. Walls are of typical frame construction with no insulation. The west wall is a duplicate of the east wall, having one window and one door in the same relative location; the south wall has neither doors nor windows. There is an attic space above and a basement space below the room, and entry to these spaces is provided by cold-storage doors. The attic, room, and basement are enclosed by a larger insulated structure, not shown in the figure, which forms corridors along the east, north, and west walls of the test room. An insulated partition wall containing cold-storage doors separates the north and west corridors; hence the west corridor can be isolated from the north and east corridors.

Direct expansion refrigerating coils in the north corridor are connected to a 12-ton capacity ammonia refrigeration plant (not shown) located on the floor below the test plant. Independently controlled banks of electric heaters are located in the corridors and in the attic space. The basement space contains both electric heaters and chilled-water coils. Fans in the basement space and the corridors aid in minimizing temperature variations.

By using the proper combination of cooling coils, electric heaters, and cold-storage doors, it is possible to operate the test room under a large variety of

simulated winter and summer conditions; for example, with either two or three "exposed walls", with temperatures of from 6 F to 130 F in the corridors, and a range of temperatures in basement and attic spaces.

Approximately 70 thermocouples are located at various points in and on the walls, floor, and ceiling of the test room and in the corridors, attic, and basement spaces. Additional thermocouples are located a few inches inside the

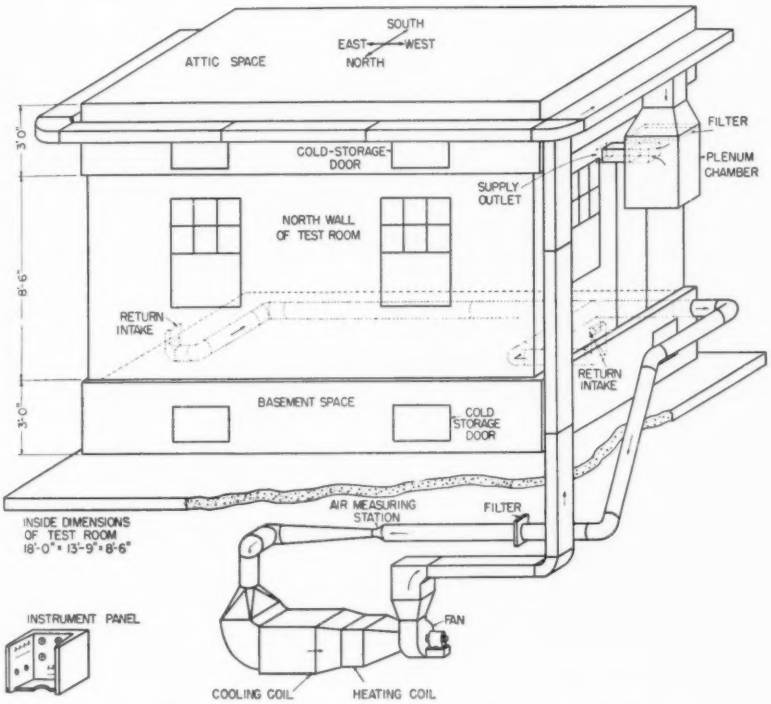


FIG. 1. DIAGRAM OF ROOM HEATING AND COOLING TEST PLANT

supply opening and return intakes. All thermocouple readings are taken at the instrument panel located on the floor below the test plant. Instrument panel equipment includes approximately 200 switches and two potentiometers—a laboratory type and an electronic self-balancing type.

Referring to the lower portion of Fig. 1, the air supplied to the test room is drawn through the cooling and heating coils by a variable speed fan that permits operation at any desired capacity between 50 and 400 cfm. After leaving the conditioning equipment the air passes through the supply duct and a filter into the plenum chamber; it then enters a 27 in. long section of horizontal duct and

discharges into the room through the supply outlet. The width and height of the horizontal duct section are the same as those of the particular supply outlet being studied. When a different size of supply outlet is to be installed, the horizontal duct section is removed and replaced by another of the proper size.

Center of the supply outlet is 1 ft below the ceiling, 8 ft from the north wall and 5¾ ft from the south wall. This location is as near the center of the west wall as construction of the room permits. Temperature of the supply air can be

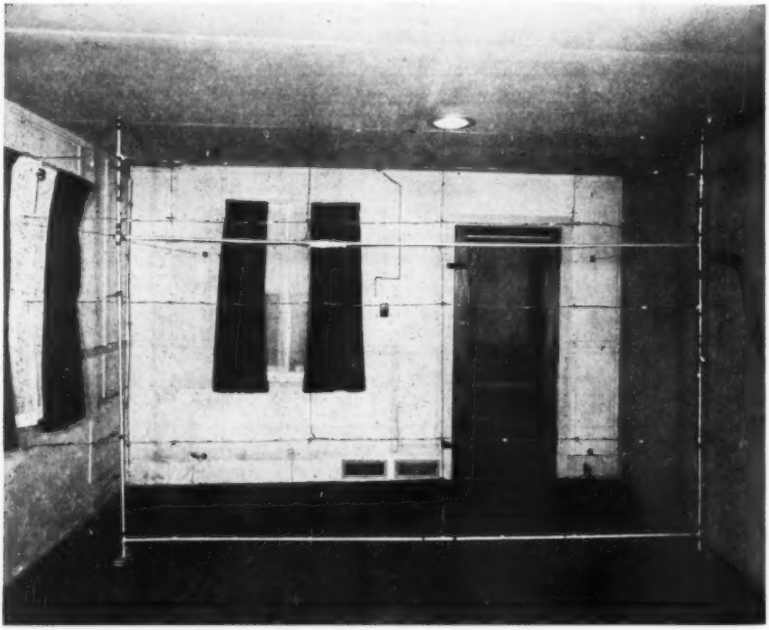


FIG. 2. VIEW OF INTERIOR OF TEST ROOM, LOOKING EAST

maintained at any desired value between 55 and 165 F. Leaving the test room, the air enters one of the 28 in. x 5 in. baseboard return intakes (the other being closed), passes through a filter in the return duct, through a nozzle to measure the air-flow rate, and re-enters the conditioning equipment.

Fig. 2 depicts part of the test room interior as viewed looking east from a position near the supply outlet. The return intake located in the east wall can be seen at approximately the center of the baseboard. The device on wheels in the foreground is a vertical rack that can be moved to any desired location along the length of the room by an electrically operated mechanism located outside the room. It supports 33 identical heated-thermocouple anemometers for measuring air velocities and air temperatures. The arrangement of the anemometers on

the rack is flexible. Leads from these anemometers pass out of the test room through flexible cable on the right of the rack and connect with switches at the instrument panel (Fig. 1), where the temperature and velocity readings are taken.

One of the heated-thermocouple anemometers with its supports is shown in Fig. 3. The anemometer was developed by the 1941 research staff and is more nearly non-directional than the one previously reported[2]. The circular-shaped object at the left is a copper sphere of ¼ in. diameter and 0.01 in. wall thickness. Inside the sphere is a tiny electric heater regulated to maintain a constant heat output. One junction of a thermocouple is embedded in the surface of the sphere;

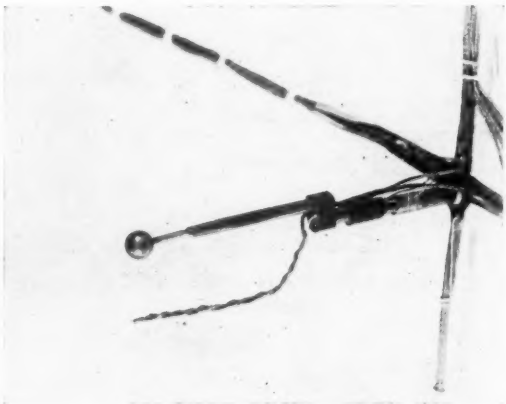


FIG. 3. PHOTOGRAPH OF HEATED-THERMOCOUPLE ANEMOMETER

this is the heated junction. The other junction is located about ¾ in. away (left end of twisted wires); this is the unheated junction. The electromotive force (emf) generated between the two junctions is primarily a function of the velocity of the air stream passing over the instrument, and the relationship for each anemometer is determined by calibration. Principal construction details and a portion of the wiring diagram are shown in Fig. 4. The wiring diagram indicates how the air temperature, as well as the velocity, is measured.

CALIBRATION OF APPARATUS

The inclined draft gages used to measure the various pressures were calibrated with a micromanometer[4] to obtain curves for correcting the gage readings to their true values.

The anemometers used in the test room were calibrated individually in a wind-tunnel[2] over a range of 20 to 2000 fpm velocity by the 1941 research staff and then installed in the room. One additional anemometer, designated the *Wind-tunnel anemometer,* was also calibrated and kept in the basement near the

wind-tunnel. When the air distribution project was reactivated in 1950, the calibrations of the nozzles in the wind-tunnel were checked and found to be unchanged. The next step was to check the calibrations of all of the anemometers; this was necessary because the emissivity of the copper spheres could have changed considerably in the nine-year period.

In order to avoid breaking the sensitive electrical connections to the anemometers mounted on the rack in the test room, recalibrating them in the wind-

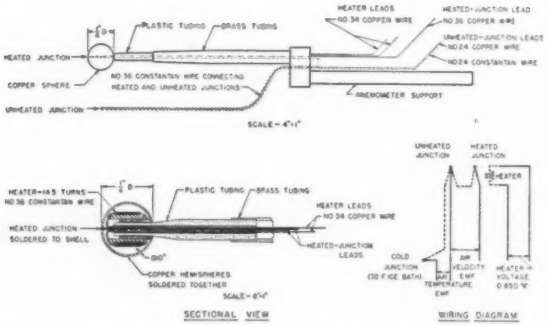


FIG. 4. PRINCIPAL DETAILS OF HEATED-THERMOCOUPLE ANEMOMETER

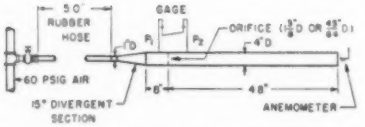


FIG. 5. ANEMOMETER CALIBRATING TUBE

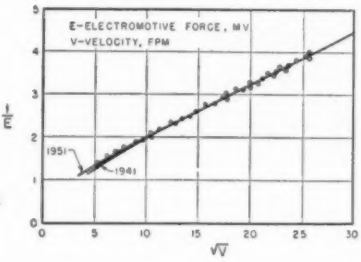
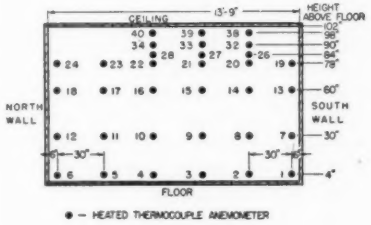
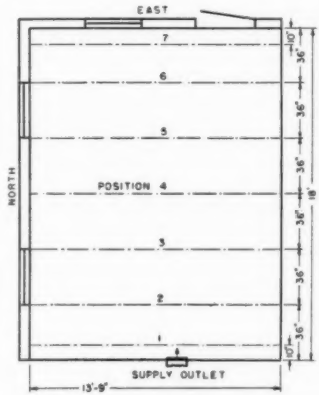


FIG. 6. CALIBRATION CURVES FOR HEATED-THERMOCOUPLE ANEMOMETERS

tunnel, and then reconnecting the electrical circuits, a means of calibrating the anemometers in place was developed. This was accomplished with a portable *calibrating tube,* the details of which are presented in Fig. 5. Air from a 60 psig source was throttled through a valve to nearly atmospheric pressure. It then flowed through 50 ft of rubber hose into a 15 deg divergent section of sheet-metal duct; thence through a thin-plate orifice in the 4-in. diameter main section of the calibrating tube. With the recalibrated wind-tunnel anemometer installed at the exit of the tube, the relationship between the static-pressure differential across the orifice and the air velocity at the exit of the tube was established. The

calibrating tube was then taken into the test room and used to calibrate each of the 33 anemometers in place on the rack. Representative calibration points and a mean curve are shown in Fig. 6. None of the more than 400 calibration points obtained for the anemometers deviated by more than 10 percent from the curve. This 1951 calibration applies over the range of air temperatures used in this investigation provided the air stream is within about 45 deg of meeting the





NOTE: Upper diagram designated 7a shows arrangement of anemometers on rack relative to interior surfaces of test room. Lower diagram designated 7b shows positions of anemometer rack along length of test room.

FIG. 7. ARRANGEMENT OF ANEMOMETERS

anemometer head-on. When the direction of the mean velocity is outside this range, corrections must be applied. The calibration curve obtained in 1941 is also presented in Fig. 6. Comparison of the 1941 and 1951 curves indicates the excellent stability of the anemometer calibration over long periods of time.

TEST CONDITIONS

Throughout the studies being reported, the arrangement of the anemometers on the rack was as shown in Fig. 7a. The average temperature at anemometers 8, 9, 10, and 11, when the rack was placed in the center of the room (position

4, Fig. 7b), was selected as the control temperature for the room conditions. This control temperature was maintained at 75 F during both winter heating and summer cooling tests.

Likewise, all studies were made with two "exposed walls"; the north and east walls. The west corridor, isolated from the north and east corridors, was maintained at approximately the test room temperature of 75 F; consequently, the west wall functioned as an "inside" wall of a room. The south wall also functioned as an inside wall, the space behind it being about 75 F at all times. Except for one test simulating an unheated crawl-space condition, the temperature in the basement space was maintained at 75 F. The single-glass windows of the test room were closed and locked. For purposes of calculating infiltration, these windows would be described as *average-fit*. Although three desk-type fans were used in the north and east corridors to minimize temperature variations, wind conditions on the external surfaces of the room, in terms of pressures and velocities, are judged to be about 2 or 3 mph.

For the heating tests under simulated winter conditions, 0 F was originally selected as the desirable air temperature in the north and east corridors. However, since 6 F was the lowest temperature which could be maintained with the refrigeration equipment, the winter heating studies were actually conducted at an "outdoor," or corridor, temperature of 6 F. Temperature in the attic space above the 4-in. batts of insulation between the ceiling joists was approximately 35 F. This temperature was regulated by setting the cold-storage doors in the north and east walls of the attic space in their wide-open positions when the corridors were at 6 F.

Indicated steady-state conditions fixed the heat loss of the test room for simulated winter conditions. The cubic feet per minute of heated supply air required to maintain a space at a desired temperature is:

$$Q_w = \frac{H_w}{(60)(0.24)d_r(t_w - t_r)} \quad \ldots\ldots\ldots\ldots \quad (1)$$

where

Q_w = flow rate of supply air, cubic feet per minute at density d_r.
H_w = heat loss of space, Btu per hour.
d_r = density of return air, pounds per cubic feet.
t_w = supply-air temperature, Fahrenheit.
t_r = return-air temperature, Fahrenheit.

As the return-air temperature in the test room was very nearly 71 F and of density 0.072 for all heating tests, these values can be substituted in Equation 1. The result is:

$$Q_w = \frac{H_w}{1.04(t_w - 71)} \quad \ldots\ldots\ldots\ldots \quad (2)$$

Since the heat loss H_w was fixed by the external conditions imposed on the test room, the selection of a value for t_w determined the flow-rate Q_w, and vice versa. The higher the selected value of t_w, the lower would be the flow rate. Therefore, the test room could be maintained at the desired temperature with either a large

flow rate of relatively low temperature supply air or a small rate of relatively high temperature air. In general, tests were conducted with flow rates of approximately 100, 230, 260, and 310 cfm, resulting in supply-air temperatures of about 143, 103, 97, and 93 F. These flow rates correspond to 3, 6½, 7¾, and 9 air changes per hr, the first being representative of winter heating practice and the last three of summer cooling practice.

Although the heat loss H_w of the room was fixed, its value was not known. However, it could be determined in two ways—either by measuring Q_w, t_w, and t_r and then using Equation 1, or by making a heat loss calculation on the test room. Since equipment was available for accurately measuring these quantities, heat loss of the test room was normally evaluated by Equation 1.

For one of the test conditions, this equation yielded a value of 7500 Btuh for H_w. For the same conditions, heat loss calculations using the HEATING VENTILATING AIR CONDITIONING GUIDE 1952[5] yielded a value of 8150 Btuh, 11 percent greater than the 7500 Btuh actual load obtained by Equation 1. The difference between the two values may be attributed to such factors as the tight construction of the test room and small safety factors in the published U and k values of materials.

In order to establish representative test conditions for the summer cooling studies, a procedure different from that used to establish the winter conditions was required, because of the lack of facilities to duplicate solar radiation effects on the room. In determining the simulated summer conditions, the test room was considered as being one room of a residence. The sensible cooling load was calculated for four different orientations of the room from THE GUIDE 1951; Chapter 12, Tables 12, 13, 17, and 25; and Chapter 2, Table 2. Design conditions used in the calculations were: 40 deg north latitude location, 95 F outdoor design dry bulb temperature, 75 F indoor dry bulb temperature, canvas awnings on the windows, and no internal heat gains with the exception of two adults seated at rest. The four orientations used and the resultant curves relating time of day to sensible cooling load are shown in Fig. 8. From these curves a value of 5000 Btuh, shown as a dotted line in the figure, was selected as a representative cooling load to impose on the test room. When this value is inserted in the equivalent form of Equation 1 for summer cooling and for air at standard density, the result is:

$$Q_s = \frac{5000}{1.08(75 - t_s)} \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

Since representative temperature differentials between the supply air and room air are 15 F and 20 F, supply air temperatures t_s of 60 and 55 F were substituted in Equation 3 to determine the flow rates for the cooling studies. The resulting two values of Q_s were 308 and 232 cfm.

Summer cooling conditions were obtained in the following manner: The electric heaters in the attic space above the insulated ceiling were set to maintain the attic temperature at 120 F. Next, the fan speed was adjusted until the flow rate Q_s was about 308 cfm as determined by the measuring station (Fig. 1). Then with the automatically controlled supply-air temperature t_s set at 60 F, the

electric heaters in the north and east corridors were adjusted until control temperature in the test room stabilized at 75 F. Three to four days of continual adjustments were required to obtain a steady-state condition. The resulting outside surface temperature of the two exposed walls was about 115 F. Since the design outdoor temperature was 95 F, the simulated outdoor temperature had to be raised considerably in order to produce effects of solar radiation on the

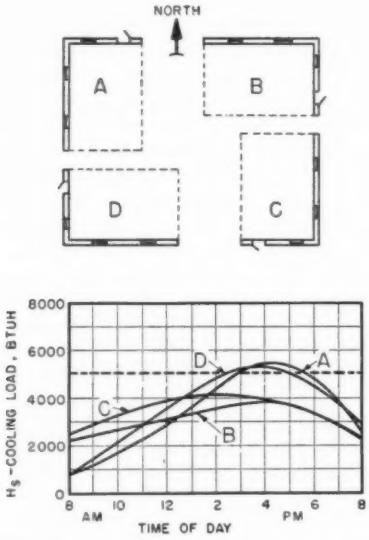


FIG. 8. RELATIONSHIP BETWEEN TIME OF DAY AND SENSIBLE COOLING LOAD OF TEST ROOM FOR FOUR ORIENTATIONS

sensible cooling load of the room. The other cooling condition using 232 cfm and 55 F supply-air temperature was obtained in a similar manner.

The essential difference between the methods used in establishing winter and summer conditions is the following: For winter heating, representative outdoor conditions were imposed on the test room and the resulting actual heating load was measured; whereas for summer cooling, a representative cooling load was selected using Fig. 8 and the outdoor conditions (north and east corridor temperatures) were adjusted until the selected cooling load was obtained.

GENERAL PROCEDURE

At the beginning of each winter heating and summer cooling test the anemometer rack was placed in the center of the room in position 4; this position, as well as others used during each test, is shown in Fig. 7b. Desired test condi-

tions were then imposed on the room and adjustments made until the desired steady state was obtained. Temperature emf and velocity emf at each of the 33 stations on the anemometer rack were then read on the electronic potentiometer at the instrument panel, and the temperatures and velocities obtained from calibration curves or tables. Next, the anemometer rack was moved to position 1 where another traverse of the temperatures and velocities was made. Since the rack was moved by externally controlled equipment, entrance into the room was not required during this phase of the test. Similar traverses were

TABLE 1—STUDIES WITH FREE OPENINGS AS SUPPLY OUTLETS[a]

TEST NO.[b]	HEIGHT OF FREE OPENING[c], INCHES	ROOM AIR TEMP, F	SUPPLY AIR TEMP, F	TEMP DIFFERENTIAL, F	OUTLET VELOCITY, fpm	RETURN INTAKE LOCATION[d]	COOLING LOAD, BTUH
1	5	76	65	11	770	West	4300
2	5	83	83	0	770	West	0
3	5	79	79	0	770	West	0
4	5	76	65	11	770	West	4400
5	5	77	65	12	770	East	4600
6	10	80	65	15	390	East	5700
7	10	79	65	14	390	West	5700
8	5	81	65	16	770	East	6100
9	5	80	65	15	770	West	5700
10	3¼	80	65	15	1230	West	5900
11	3¼	80	65	15	1230	East	5900
12	4	79	65	14	960	East	5700
13	4	80	65	15	960	West	5900
14	6½	79	65	14	590	West	5600
15	6½	80	65	15	590	East	5800
16	6½	80	65	15	590	West	5600
17	6½	80	65	15	590	East	5800
18	6½	72	72	0	590	West	0
19	6½	75	75	0	590	East	0
20	6½	79	65	14	590	West	5200
21	6½	80	65	15	590	West	5900

[a] Flow rate for all tests: 380 cfm (10.8 air changes per hour).
[b] Tests Nos. 1-15 conducted in 1941-42.
[c] Width of free opening is 14 in.
[d] See Fig. 1 for supply outlet and return intake locations.

made at positions 2, 3, 5, 6 and 7. Finally, the rack was again moved to position 4, and a check made to insure that conditions had not changed during the four-hour period required to make the traverses.

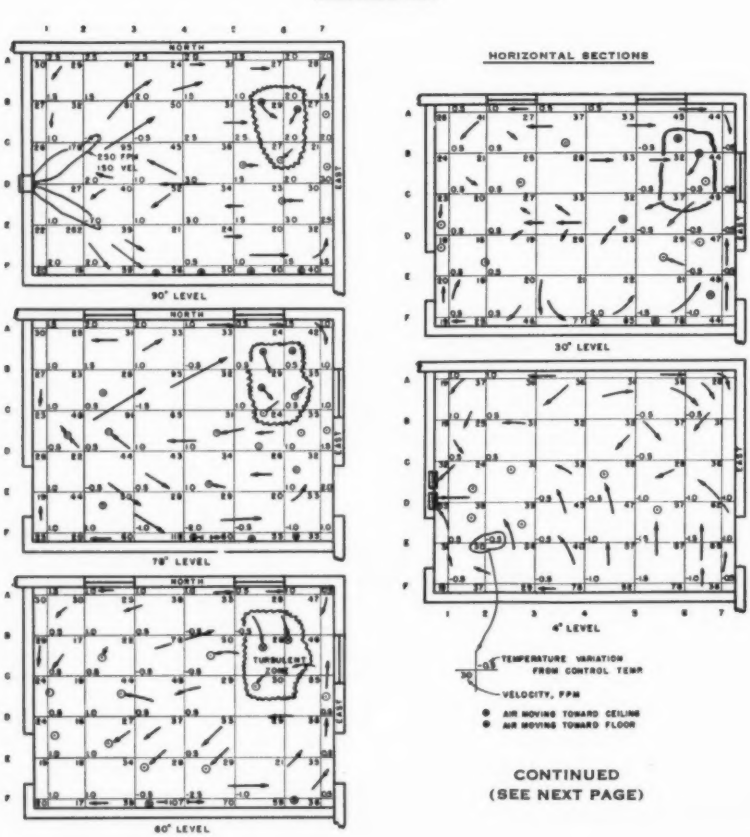
Investigators then entered the room with smoke apparatus and a commercial anemometer of the hot-wire type. Smoke was used to determine the direction of the lower velocity air currents at numerous points in the room, and also the general motion of the air masses in the room. The commercial anemometer was used to measure the air velocity at numerous points in the occupied zone, as well as to determine the jet characteristics close to the supply outlet and at some distance away. The portion of the test conducted with investigators in the room required about three hours.

Using this procedure, tests were conducted under both winter heating and summer cooling conditions with various sizes of supply outlets ranging from

14 x 10 in. to 14 x 4 in. and with air-flow rates between 105 and 380 cfm. The following types of supply outlets were used: free openings (holes in the wall), commercial shallow-vane and deep-vane grilles, and commercial shallow-vane registers. The shallow vanes were about $\frac{3}{8}$ in. deep and the deep-vanes about $\frac{3}{4}$ in. deep. Each supply outlet was located high in the center of the west wall as shown in Fig. 1, and the return intake was located in the baseboard of either the west or east wall.

FREE OPENINGS—RESULTS AND CONCLUSIONS

Studies with free openings as supply outlets are described in Table 1. The first 15 tests were conducted in 1941-42, but have not been reported previously.



CONTINUED
(SEE NEXT PAGE)

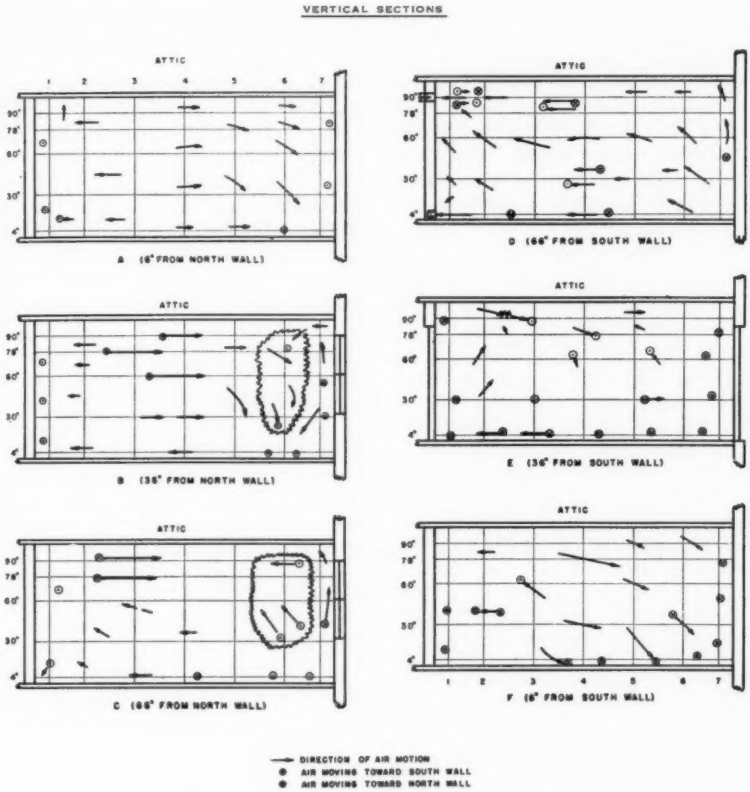


FIG. 9. EXAMPLE OF MANNER OF COMPILING TEST DATA

Since the test conditions used at that time differ somewhat from those currently in use, the discussion of test conditions given in a previous section of this paper does not apply to these.

Fig. 9 exemplifies the method used to compile the principal data obtained for each test. Such diagrams, combined with supplementary data, served as the basis for analyzing and interpreting the data from all tests. The vertical lines numbered 1 through 7 refer to the seven positions of the anemometer rack in the room (Fig. 7b). The left side of the figure shows five horizontal sections of the room at the 90, 78, 60, 30, and 4-in. levels. The horizontal lines, lettered *A* through *F*, along the left edge refer to the locations of vertical columns of anemometers relative to the north and south walls. Referring to Fig. 7a, as the rack is moved the column comprised of anemometers 6, 12, 18 and 24 travels

along line *A*; and the column comprised of anemometers 1, 7, 13, and 19 travels along line *F*. At each intersection of a horizontal and a vertical line in the diagrams on the left side of Fig. 9, the upper number indicates the temperature variation from the control (room) temperature and the lower number indicates the velocity—for example, at the 4-in. level at *E-2*, the temperature is 0.5 deg below the control temperature of 75 F, and the velocity is 30 fpm. Where no temperature value is given, the deviation is zero.

The right side of Fig. 9 shows six vertical sections of the room. The top diagram is a vertical section lying along the lines marked *A* in the diagrams on the left side of the figure. If these six vertical sections were cut out of the page and fixed in upright positions, and then if the five horizontal section views at the left were sandwiched into these vertical sections in an egg-crate manner, a three dimensional representation of the temperature variations and velocities at 231 points in the room would be obtained.

The principal purpose of Tests Nos. 1-5 of Table 1 was to gain familiarity in operation of the test plant and to develop a procedure for conducting tests and compiling the data. The objective of Nos. 6-15 was to determine the effect of varying the outlet velocity and the return intake location. Desired velocity variation was accomplished by varying the size of the free opening. For these tests the room temperature, supply-air temperature, flow rate, and cooling load were essentially constant. If the relatively small change in aspect ratio of the opening is ignored, the only variable in Nos. 6, 8, 11, 12, and 15 is velocity, the return intake location being fixed at the east wall position. Results of these five tests were compared to evaluate the effect of velocity on the air distribution. This applies to Nos. 7, 9, 10, 13, and 14, which were conducted with the return intake located in the west wall. The effect of the intake location was then evaluated by comparing the results of No. 6 with No. 7, No. 8 with No. 9, and so on.

Results with velocity as the only variable showed that an increase in outlet velocity caused the room air motion to increase and the temperature variations in the occupied zone (floor to 6 ft above floor level) to decrease. Therefore, with low outlet velocities, the problem is primarily that of a low velocity stream of cold air dropping into the occupied zone. At high outlet velocities, the problem is primarily that of high velocity currents of air at about room temperature existing in one or more regions of the occupied zone. This indicates there is an optimum value of outlet velocity at which temperature variations will be small and room air motion not excessive. For these studies with free openings, such an optimum velocity appeared to be of the order of 800 fpm.

Results with the location of the return intake as the only variable showed that the return intakes had only a very localized effect on the air distribution. Low velocity air currents within about 1 ft of the intake were drawn into it; air currents farther away than this were unaffected except in the region of the room below the 4-in. level. In this region near the floor surface, the east return caused a gentle flow of air eastward, whereas the west return caused a gentle flow westward. In both cases, the scalar magnitude of the air velocity; *i.e.*, the speed, was about the same at corresponding points in the region; hence, in this region the effect of the returns was only to change the direction of the velocity. Therefore, the location of the return intakes had negligible effect on the air distribution in the test room.

The general pattern of the air distribution was the same for all of tests Nos. 6-15. Horizontal spread of the jet was about 20 deg throughout its travel along the length of the room. With the highest outlet velocity, the lower edge of the jet entered the occupied zone at a point about three fourths of the room length or about 13 ft from the supply outlet; with the lowest velocity, it entered the occupied zone at about the center of the room or 9 ft from the outlet. The upper edge of the jet remained close to the ceiling at least until it reached the middle of the room. As the jet velocity decreased with distance from the outlet, the gravitational forces due to temperature differences between the jet air and the room air became predominant, with the result that the jet axis curved downward and into the occupied zone. For all tests, even those with an outlet velocity as low as 390 fpm, at least the upper portion of the jet struck the east wall; *i.e.*, overblow occurred, indicating that the jet induced a relatively small quantity of room air.

Except in the regions where the jet entered the occupied zone, temperature variations from the control temperature were extremely small, seldom being greater than 1 deg. Consequently, vertical temperature variations in the occupied zone, which are often large in winter heating applications, do not appear to be a problem during summer cooling.

For the first two tests conducted in 1951, Nos. 6 and 17 in Table 1, conditions duplicating those of Nos. 14 and 15 were used. The results showed temperatures and velocities at the various points in the room were in good agreement with the 1941 tests, indicating that test conditions and data could be duplicated.

The isothermal studies, Nos. 18 and 19, showed characteristics, except for drop, very similar to those of the cooling tests, Nos. 16 and 17, in which a 15 deg differential was used. This suggests the possibility that data on performance of various types of supply outlets obtained by other investigators, notably by those at Case Institute of Technology, can be extended to summer cooling applications utilizing small temperature differentials.

Tests Nos. 20 and 21 were special studies conducted to determine the effect of imposing various proportions of the total cooling load on different room surfaces. During No. 20 about 85 percent of the load was applied on the exposed walls and none through the floor, whereas in No. 21 about 60 percent was applied through the floor of the room by heating the basement space. The results of these studies were compared with No. 16, in which 56 percent of the load was applied through the ceiling.

The jet axis in No. 16 remained in a vertical plane perpendicular to the east wall, whereas in No. 20 the jet did not drop as much and the jet axis curved toward the south wall. For No. 21, the jet dropped even less, the flow patterns in the room were significantly different, and the jet axis curved even more toward the south wall. These effects were apparently due to free convection currents along the room surfaces. The higher north corridor temperature of 138 F in No. 20, as compared to 103 F in No. 16, set up stronger free convection currents in a clockwise circulation (looking from the supply outlet) about the room. Since the jet initially issues from an opening near the ceiling, the free convection currents moving along the ceiling surface in a north to south direction tend to push the jet toward the south wall. When the average basement

temperature was 134 F and the north corridor temperature 108 F (No. 21), the free convection currents were apparently even stronger than during No. 20, because the jet moved still farther toward the south wall. This signifies that free convection effects at the various surfaces of a room can have definite influence on the air distribution.

TABLE 2—STUDIES WITH SHALLOW- AND DEEP-VANED SUPPLY OUTLETS

Test No.	Outlet Type	Height of Outlet[b] Inches	Vertical Vane Setting	Horizontal Vane Setting	Room Air Temp F	Supply Air Temp, F	Temp Differential, F	Outlet Velocity, fpm	Return Intake Location[c]	Flow Rate, cfm	Air Changes Per Hr	Load, Cooling or Heating, Btuh
22		8	*A*	—	76	60	16	600	West	315	9	4800
23		8	*A*	—	75	60	15	600	East	315	9	4500
24		8	*A*	—	74	55	19	430	West	230	6½	4300
25		8	*A*	—	74	55	19	430	East	230	6½	4200
26		8	*B*	—	75	55	20	410	West	230	6½	4400
27		8	*B*	—	75	60	15	580	West	315	9	4700
28		8	*C*	19° up	75	60	15	560	West	315	9	4700
29		8	*D*	28° up	76	56	20	410	West	230	6½	4700
30	Shallow	8	*E*	—	78	59	19	420	West	260	7¾	5000
31	Vane	6	*E*	—	78	58	20	570	West	250	7¾	5100
32		5	*E*	—	79	58	21	680	West	260	7¾	5400
33		4	*E*	—	79	58	21	980	West	260	7¾	5400
34[a]		8	*C*	0°	76	95	19	600	West	315	9	7100
35[a]		8	*F*	10° Down	75	97	22	520	West	270	7¾	7200
36[a]		8	*C*	0°	75	103	28	440	West	225	6½	7500
37[a]		8	*C*	0°	75	143	68	220	West	105	3	7800
38[a]		8	*F*	10° Down	74	97	23	510	West	265	7¾	7700
39		8	*C*	0°	84	84	0	580	West	315	9	0
40		8	*G*	—	75	57	18	430	West	230	6½	4200
41	Deep	8	*G*	—	75	55	20	430	East	230	6½	4200
42[a]	Vane	8	*G*	—	75	103	28	470	East	230	6½	7700
43[a]		8	*G*	—	75	104	29	470	West	230	6½	8000

a Winter Heating Tests.
b Width of Outlet 14 in.
c See Fig. 10 for description of vertical vane settings and Fig. 1 for supply outlet and return intake locations.

These studies show that free openings of the sizes tested are unsatisfactory for summer cooling installations, principally because of the small amount of induction and the lack of control over the jet with subsequent overblow and excessive drop.

VANED OUTLETS—RESULTS AND CONCLUSIONS

The studies made with vaned outlets are presented in Table 2. Of those listed, 14 are summer cooling and 7 are winter heating studies, and one is an isothermal exploration. The studies encompass two types of vaned outlets, seven vertical and four horizontal vane settings, two different supply-air tempera-

tures for cooling and four for heating, two return intake locations, and four rates of flow.

The vertical vane settings in column 4 of Table 2 are described in Fig. 10. The vanes of the commercial supply outlets used with settings *A*, *B*, *C*, *D*, and *F* could be adjusted to any desired angle in groups of 6 or 7; whereas those of setting *E* were fixed at 22 deg right and left deflections. The vanes of the

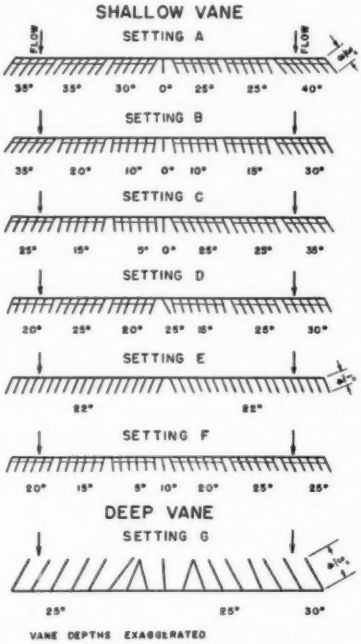


FIG. 10. DESCRIPTION OF VERTICAL VANE SETTINGS

deep-vane outlet, setting *G*, could be individually adjusted. Where horizontal settings are not given in column 5 of the table, the supply outlet was not equipped with horizontal vanes; *i.e.*, it was a single-deflection type. The location of all supply outlets is shown in Figs. 1 and 7b.

A study of the effect of the return intake location during summer cooling conditions (Tests Nos. 22-25, 40, and 41) showed, as with the free openings, that the intake location had negligible effect on the air distribution in the room. Tests Nos. 42 and 43, conducted under winter heating conditions and with the return intake location as the only variable, showed the same trend. However,

these two tests alone are not considered sufficient to substantiate a definite conclusion that the intake had negligible effect on the air distribution during winter heating. All of these tests showed that the particular vane setting of an outlet had great influence on the air distribution whereas the vane depth did not. The remaining 14 tests were therefore limited to one return intake location and one outlet type.

In studying the effect of both the horizontal and vertical vane settings on the air distribution, velocity and temperature probes were conducted in the room prior to each test and the vanes of the supply outlet adjusted until the air distribution reached an apparent optimum. Vertical vane setting *A*, the first one used, Test 22, was a result of attempts to prevent excessive drop of the jet by increasing the spread. Although this setting eliminated the problem of drop, the supply air was split into two distinct jets which impinged on the adjacent walls at points about 45 deg from the outlet, one jet striking the north wall and one the south wall. A stream of air then flowed eastward along each of these walls and to the center of the east wall; here the two streams met and formed one stream that traveled up the center of the room in the direction of the supply outlet. The result was a general movement of air in a clockwise direction, looking from above, in the north half of the room and a counterclockwise movement in the south half.

In order to eliminate the undesirable impingement on the adjacent walls, further experimentation with the vanes was conducted, the result being setting *B*. This setting also produced two jets. The jet directed at the north wall turned away from the wall and back toward the center of the room; the jet directed at the south wall hit that wall and continued along it. Consequently, a counterclockwise movement of the air was set up throughout the room.

Further explorations led to setting *C*, which directed more air toward the north wall. In contrast to the effect of setting *B*, a clockwise movement of air throughout the room resulted with setting *C*. Since the air distribution in the room was significantly different for each of the three vane settings, the vertical vane setting of a high sidewall supply outlet is evidently an extremely important factor affecting the air distribution in a room during cooling.

The most uniform temperatures and velocities in the occupied zone during cooling were obtained with vertical vane setting *C* and a 19 deg upward deflection of the air with horizontal vanes located behind the vertical vanes (Test No. 28). This test was conducted with a 15 deg temperature differential. When a 20 deg differential was used, excessive drop occurred and further experimentation with the vertical and horizontal vane settings was required. Although vertical vane setting *D* was eventually selected and Test No. 29 conducted, no combination of settings of the horizontal and vertical vanes was found which completely prevented cold air from entering the occupied zone. This indicates that, when conventional high sidewall supply outlets are used in residences, air distribution problems can be expected to become increasingly difficult as supply-air temperature is decreased below that corresponding to a 15 deg differential.

The objective of Nos. 30-33, which were summer cooling tests, was to study the effect of outlet velocity. As with free openings, the room air motion increased as the outlet velocity increased. In addition, air noise at the outlet

was clearly audible at the outlet velocity conditions of 680 and 980 fpm in Nos. 32 and 33. Instead of the single jet expected with these conventional 22 deg right and left deflection outlets, setting *E*, Fig. 10, two jets were produced. This two-jet condition was not evident by a velocity traverse taken within about 1 in. of the outlet, since such traverses always indicated a uniform velocity across the outlet face. However, within a very few inches of the face, the air issuing from about the center half of the outlet split into two streams, one of which was deflected sharply to the right and the other sharply to the left. These streams joined the air issuing from the two outside portions of the outlet and formed two distinct jets. Moreover, the included angles of the axes of the two jets was *greater* than the 44 deg included angle of the vane settings, this is in contradiction with information available in current technical literature[5]. These effects are believed to be caused by the existence of a pressure gradient in the jet close to the outlet; further analytical and experimental work on these effects is contemplated.

Explorations showed that a low-pressure region existed close to the supply outlet and between the two jets, and that room air would be drawn into this region. Along an axis through the center of the outlet, and normal to the outlet face, room air flowed directly toward the supply outlet and to within about three inches of the face before being induced by one of the jets and carried away.

These effects: the two jets, their greater than expected included angle, and the flow of room air to within a few inches of the outlet face are not peculiar characteristics of the test plant. The same effects were observed in two research residences being operated under summer cooling conditions representative of usual residential practice.

The primary purpose of the winter heating studies was to determine whether the relatively high flow rates employed during cooling could be utilized successfully during winter heating. Tests Nos. 34 and 36 were therefore conducted with the cooling flow rates of 310 and 225 cfm. Results were compared with those of a test, No. 37, utilizing only 105 cfm, which is representative of current heating practice for such a room.

For all heating tests of Table 2, gravitational forces due to the difference in densities of the supply air and room air were found to be extremely important. In each case the jet immediately rose toward the ceiling and practically all of the induction and air motion took place above the occupied zone; consequently, the air velocities in the occupied zone were generally very low. With the highest flow rate of 315 cfm, the velocities at numerous points in the occupied zone were only about 30 fpm; with the lowest flow rate of 105 cfm, they were of the order of 15 fpm. When smoke studies were made in the room during the 105 cfm test, a thin cloud of smoke remained practically stationary in a region at the 60-in. level. With the 230 cfm tests, this same sort of stagnation region existed at the 30 in. level. The minimum flow rate at which a stagnation region did not occur was found to be 270 cfm; Test 35 was therefore conducted with this flow rate. During exploratory studies and Tests Nos. 35 and 38, attempts to increase the air motion, and thereby reduce vertical temperature variations in the occupied zone, by directing the supply-air downward at various angles were unsuccessful. Although a horizontal vane setting such as 10 deg down would initially direct the jet toward the floor, within a very few feet of travel from

the outlet it would rise above the occupied zone. In fact, it was as difficult to get the warm supply-air down into the occupied zone during heating as it was to prevent the cool supply air from dropping into the occupied zone during cooling.

During Test No. 37, with a flow rate of 105 cfm and a corresponding supply air temperature of 143 F, the average vertical temperature variation between the 4-in. and 78-in. levels in the room was 23 F. Increasing the flow rate to 225 and then 315 cfm (Tests Nos. 36 and 34), reduced the variations to 17 and 11 F deg, respectively (the supply-air temperature was 103 F during No. 36 and 95 F during No. 34). Thus, high rates of flow during winter heating are advantageous in reducing temperature variations in a room. Moreover, comparison of the heating loads given in Table 1 for these three tests shows that the load decreases as the supply-air temperature decreases. This is attributed to the lower temperatures existing in the upper portions of the room with the smaller temperature variations. This effect on heating loads has been reported by other investigators.[6] These favorable characteristics of high flow rates; namely, reduction of temperature variations and a decrease in heating load, together with the fact that the room air motion was not excessive even at a flow rate as high as 315 cfm, indicates excellent possibilities of year 'round air conditioning systems successfully utilizing the same flow rate during both winter heating and summer cooling.

The conditions of a special heating test, No. 38, were similar to those of No. 35 except that the temperature of the basement space was maintained at 46 F during the former and at 75 F during the latter. As a consequence, the floor surface temperature during No. 38 was correspondingly lower than during No. 35. With the simulated *crawl-space* condition of 46 F, the previously described stagnation region existed in the 30 in. level, whereas no such stagnation region existed in the test room with the simulated *heated-basement* condition of 75 F. This indicates that air distribution problems can be expected to become more difficult as the temperature of the space beneath the floor of a room is lowered, with consequent lowering of the floor surface temperature.

SUMMARY OF RESULTS AND CONCLUSIONS

1. The free openings were unsatisfactory as high sidewall supply outlets for summer cooling.

2. During the summer cooling studies, the location of the return intake had only a localized effect on the air distribution when both free openings and vaned outlets were used. The same trend was found in the limited number of winter heating studies conducted.

3. At the same outlet velocity, the air motion in the occupied zone was much greater during the cooling studies than during the heating studies.

4. Vertical temperature variations in the occupied zone during cooling were generally about 1 deg, whereas they ranged between 10 and 20 deg during heating.

5. Vertical-vane settings of the supply outlets had a great influence on the air distribution in the occupied zone during cooling, but only a small influence during heating.

6. Horizontal-vane settings of the supply outlets affected the air distribution in the occupied zone during cooling, but did not affect it during heating.

7. Gravitational forces due to the difference in the densities of the supply air and the room air had an extremely important effect on the air distribution during both cooling and heating.

8. An increase in the flow rate during winter heating, with accompanying decrease in supply-air temperature, caused a decrease in the temperature variations within the room and a reduction of the room heat loss, but did not cause excessive air motion in the occupied zone. This indicates excellent possibilities of year 'round air conditioning systems successfully utilizing the same flow rate during both winter heating and summer cooling.

ACKNOWLEDGMENT

This paper is a report of one phase of a research program being conducted under the terms of a cooperative research agreement between the *American Gas Association* (PAR Plan, *AGA* Project DGR-2-AC) and the Engineering Experiment Station of the University of Illinois. The broad objective of this research program, initiated in October 1947, is the improvement of air distribution systems for year 'round gas air conditioning. Results of research previously completed under the terms of this agreement have been reported[1, 7-11]. Acknowledgment is made for the advice and counsel rendered by the *A.G.A.* Technical Advisory Group for Heating and Air Conditioning Research, and especially for the help and suggestions by W. F. Friend, H. C. Pierce, and G. F. Zellhoefer, Technical Advisors for the project.

Acknowledgment is also made to H. A. Larson and W. G. Zimmerman, former research assistants in mechanical engineering, for their assistance in collecting and compiling the data. The material presented in this paper will ultimately comprise part of a University of Illinois Engineering Experiment Station Bulletin.

REFERENCES

1. *Improvement of Air Distribution Systems for All-Year Air Conditioning,* by S. F. Gilman, R. J. Martin and S. Konzo (*American Gas Association,* New York, N. Y., First Edition, 1950).

2. Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, pp. 351-66).

3. Effect of Room Dimensions on the Performance of Direct Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock, E. L. Broderick and S. Sachs (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, pp. 331-46).

4. Investigation of Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz and V. S. Day (*University of Illinois Engineering Experiment Station Bulletin* No. 120, 1921, pp. 91-101).

5. HEATING VENTILATING AIR CONDITIONING GUIDE 1952 (Published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.).

6. Performance of a Gas-Fired Forced-Air Heating System in Research Residence No. 1, by S. Konzo, G. H. Green, R. W. Roose, and M. E. Childs (*University of Illinois Engineering Experiment Station Bulletin* No. 397, 1951).

7. Pressure Loss and Air Flow Characteristics of a Box Plenum, by S. F. Gilman, R. J. Martin, W. R. Hedrick, and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, pp. 299-320).

8. Investigation of the Pressure Characteristics and Air Distribution in Box-Type Plenums for Air Conditioning Duct Systems, by S. F. Gilman, R. J. Martin and S. Konzo (*University of Illinois Engineering Experiment Station Bulletin* No. 393, 1951).

9. *Manual for the Design of Box-Plenum Type Air Conditioning Duct Systems* (*American Gas Association,* New York, N. Y., 1952).

10. *Progress in Air Distribution for Gas All-Year Air Conditioning,* by S. F. Gilman (Paper presented at Pacific Coast Research and Utilization Conference, Berkeley, Calif., April 1951).

11. Pressure Losses of Take-Offs for Extended-Plenum Duct Systems, by J. W. Holl, S. F. Gilman, R. J. Martin and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, pp. 419-36).

1472

No. 1472

FIELD STUDIES OF FLOOR PANEL CONTROL SYSTEMS

By A. B. ALGREN*, E. F. SNYDER, JR.**, AND J. S. LOCKE†,
MINNEAPOLIS, MINN.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the University of Minnesota, Minneapolis, Minn.

THE PURPOSE of the research program under which the studies on floor panels were made, is to determine, from actual investigation in the field, the optimum method of control for each basic type of heating panel installed in various types of structures having varying percentages of glass area. The work involved in this program came under the general direction of the Committee on Research and under the direct control of Group D of the Technical Advisory Committee on Panel Heating and Cooling‡.

Since the effectiveness of a control system is based largely on the conditions maintained within the controlled space and since desirable comfort conditions for spaces heated by panel heating systems have not as yet been outlined by Group C of the T.A.C. on Panel Heating and Cooling, the criterion selected for this study was the *stability* of the controlled temperatures. The stability of a control point is not to be interpreted as meaning constant temperature, but rather a temperature which may vary in some relationship with load conditions including such factors as: (1) outdoor dry bulb temperature; (2) solar effects; (3) wind effects; (4) building construction (glass area, infiltration, trans-

---

* Professor, Department of Mechanical Engineering, University of Minnesota. Member of A.S.H.V.E.

** Supervisor, Products Application, Commercial Division, Minneapolis-Honeywell Regulator Co. Member of A.S.H.V.E.

† Sales Manager, Commercial Division, Minneapolis-Honeywell Regulator Co. Member of A.S.H.V.E.

‡ Personnel—Group D: E. F. Snyder, Jr., *chairman;* A. B. Algren, *vice chairman;* H. W. Alyea, Louis Barfus, S. D. Browne, R. L. Campbell, C. A. Gustafson, W. J. Hajek, H. H. Kieckhefer, W. H. Kliever, H. T. Kucera, J. S. Locke, J. F. McCauley, P. F. Neess, L. C. Plaehn, J. K. M. Pryke, A. O. Roche, Jr., C. W. Signor.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January, 1953.

mission effects, and storage capacity); and (5) occupancy effects. It is felt that if stability can be realized then any comfort pattern subsequently specified by Group C can be obtained.

The magnitude of such an investigation as outlined is evident. In order to obtain basic information which would be of immediate value to the profession, it was therefore decided to limit this study to the following types of construction:

Type A—Heavy floor panel, heavy building construction, large glass area.
Type B—Heavy floor panel, light building construction, large glass area.
Type C—Heavy floor panel, light building construction, small glass area.

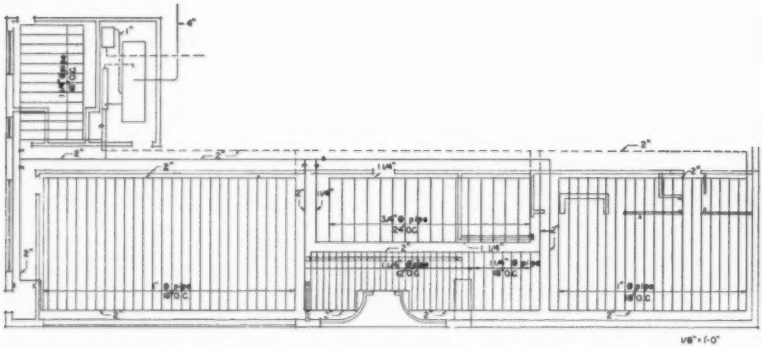


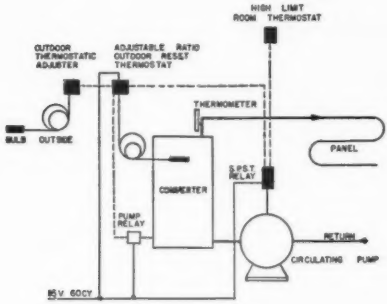
FIG. 1. HEATING SYSTEM PIPING DIAGRAM FOR INSTALLATION A

TYPE *A* INSTALLATION

The installation under study for Type *A* construction is a general office in a factory building located in Milwaukee. This office section is a single story, 124 ft long, 24 ft wide with a ceiling height of 10 ft. The long dimensions face north and south, the north wall consists of 75 percent glass, while the south wall is an interior partition wall. The east and west walls have no glass area. Wall construction is 8-in. cement block and 4-in. face brick with plaster for the interior finish, 2 in. of fill insulation with a 2 ft air space between the ceiling and roof. The floor or panel construction consists of 6 in. of concrete laid on a sand and gravel fill, with rubber tile as the floor covering. The windows are double glazed and weather-stripped.

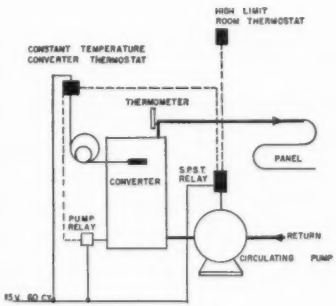
The floor panel heating system consists of five grid type panels buried to a depth of 3 in. in the concrete. The piping layout is shown in Fig. 1. The pipe is welded wrought iron and varies in size and spacing as follows:

Northeast panel section, 1-in. pipe on 18-in. centers.
Northwest panel section, 1-in. pipe on 18-in. centers.
Center panel section (North), 1¼-in. pipe on 12-in. centers.
Center panel section (South), ¾-in. pipe on 24-in. centers.
East panel section, 1¼-in. pipe on 18-in. centers.
All headers are 2-in. pipes.



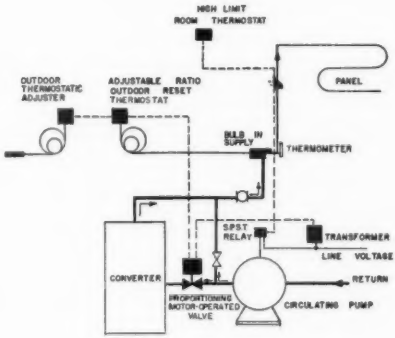
SYSTEM NO. 1

ROOM THERMOSTAT CONTROLS CIRCULATING PUMP.
WATER TEMPERATURE VARIABLE FROM OUTSIDE
ADJUSTER BY CONTROLLING CONVERTER PUMP.



SYSTEM NO. 2

ROOM THERMOSTAT CONTROLS PANEL PUMP.
CONVERTER WATER TEMPERATURE CONSTANT



SYSTEM NO. 3

PROPORTIONS VALVE FROM OUTSIDE ADJUSTER
PANEL WATER TEMPERATURE VARIABLE.
CONVERTER WATER TEMPERATURE CONSTANT.

FIG. 2. SCHEMATIC DIAGRAM OF CONTROL SYSTEM FOR INSTALLATION A

The source of hot water for the floor panel is a water-to-water converter to which the hot water from a steam boiler is circulated by means of a converter pump. The intermittent operation of the pump controls the temperature in the converter. A constant source of hot water is available to the converter by means of a low limit temperature controller located below the water level in the boiler, and controlling the burner so as to maintain a minimum boiler water temperature.

The control systems furnished for the 1951-1952 heating season provided the following methods of control:

System Number 1: Room thermostat controlling the panel pump. Water temperature variable from outside temperature by controlling converter pump.

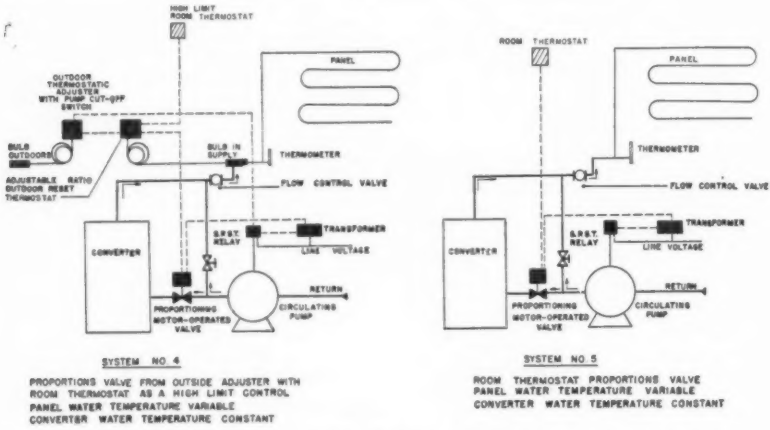


FIG. 3. SCHEMATIC DIAGRAM OF CONTROL SYSTEM FOR INSTALLATION A

System Number 2: Room thermostat controlling panel pump. Converter water temperature constant.

System Number 3: Proportioning valve controlled from outside temperature. Panel water temperature variable. Converter water temperature constant.

System Number 4: Proportioning valve from outside temperature with room thermostat as a high limit control. Panel water temperature variable. Converter water temperature constant.

System Number 5: Room thermostat controls proportioning valve. Panel water temperature variable. Converter water temperature constant.

In addition, the room thermostat controlling the panel pump can be prevented from acting as an operating control at any time by setting it to its highest temperature setting. Schematic wiring diagrams for each of these control systems are shown in Figs. 2 and 3.

TYPE *B* INSTALLATION

The installation under study for Type *B* construction is a floor panel heating system for the factory area of a second Milwaukee plant.

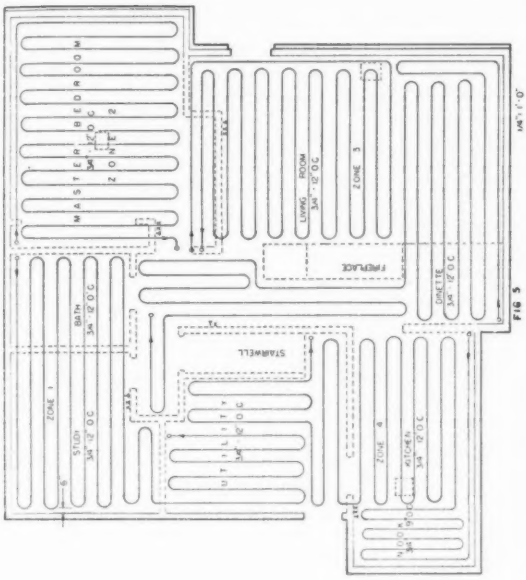


FIG. 5. MINNEAPOLIS RESIDENCE PIPING DIAGRAM FOR INSTALLATION C

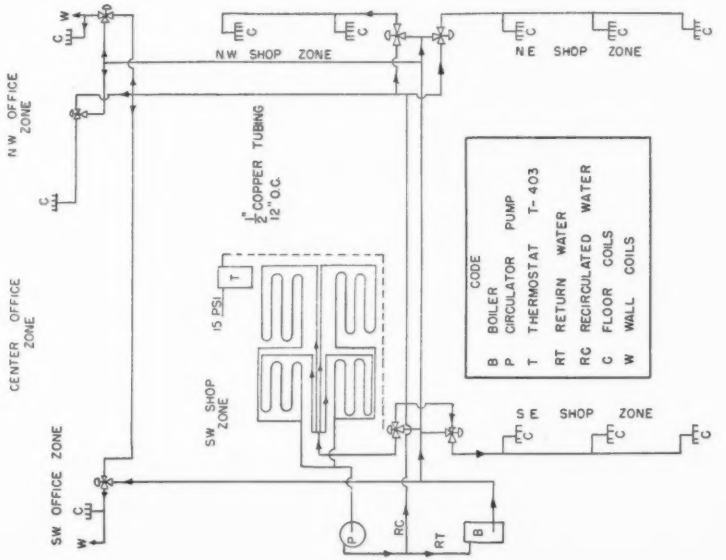


FIG. 4. SCHEMATIC RADIANT HEATING DIAGRAM FOR INSTALLATION B

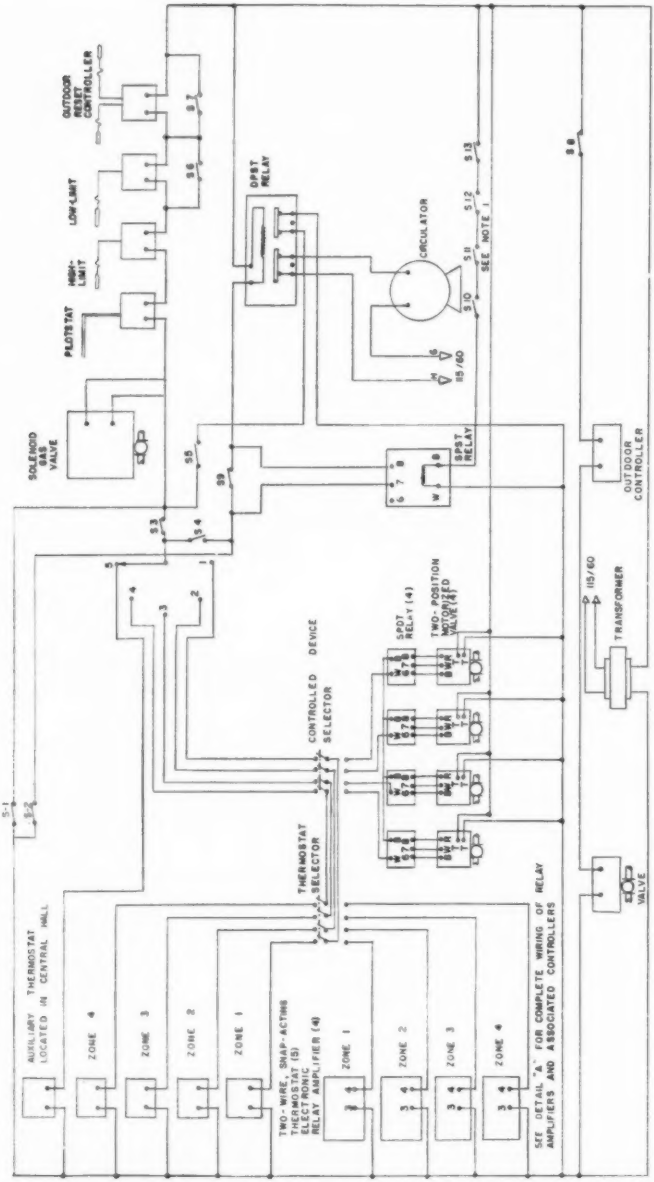
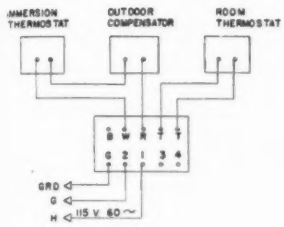


FIG. 6. MINNEAPOLIS RESIDENCE WIRING DIAGRAM FOR INSTALLATION C

(Continued on Next Page)



NOTES:

1. SWITCHES S10 – S13 ARE INSTALLED ON THE MOTOR SHAFTS OF THE MOTORIZED VALVE AND EACH SWITCH IS CLOSED WHEN THE ASSOCIATED VALVE IS CLOSED

2. SWITCHES S1 – S9 ARE OPERATED BY THE CONTROL – MODE SELECTOR SWITCHES IN ACCORDANCE WITH THE SWITCH SCHEDULE

SWITCH SCHEDULE

MODE OF CONTROL	WATER TEMP	SWITCH ACTION S1	S2	S3	S4	S5	S6	S7	S8	S9
VALVE CONTROL-CONTINUOUS CIRCULATOR	FIXED	CL	CL	OP	OP	OP	OP	CL	OP	OP
VALVE CONTROL-CONTINUOUS CIRCULATOR	RESET	CL	CL	OP	OP	OP	CL	OP	OP	OP
CIRCULATOR CONTROL	FIXED	CL	OP	OP	CL	OP	OP	CL	OP	CL
CIRCULATOR CONTROL	RESET	CL	OP	OP	CL	OP	CL	OP	OP	CL
BURNER CONTROL-CONTINUOUS CIRCULATOR		OP	CL	CL	OP	OP	CL	CL	OP	CL
BURNER & CIRCULATOR CONTROL		OP	OP	OP	CL	CL	CL	CL	OP	CL
CONTINUOUS CIRCULATOR CONTROL	FIXED	CL	OP	OP	CL	OP	OP	CL	CL	CL

IDENTIFICATION OF ZONES
ZONE NO 1 - DEN
ZONE NO 2 - BEDROOM
ZONE NO 3 - LIVING - ROOM
ZONE NO 4 KITCHEN & UTILITY
SCHEMATIC CIRCUIT
HEATING CONTROL INSTALLATION

FIG. 6 *(Continued)*. MINNEAPOLIS RESIDENCE WIRING DIAGRAM FOR INSTALLATION C

This is a single story building, 247 ft long, 92 ft wide with a ceiling height of 18 ft. The long dimension faces north and south. The north and south walls each have approximately 46 percent glass area, the east wall 50 percent glass area, while the west wall is an interior partition wall. Wall construction is 8 in. of common brick and 4 in. of face brick. Ceiling construction consists of 1-½ in. of wood with 1 in. rigid insulation. Floor construction is 6 in. of reinforced concrete laid on 3 ft of gravel fill with no floor surface covering. Windows are single glazed industrial type glass with no weather stripping.

The floor panel heating system for each zone consists of ½-in. copper tubing, spaced on 12-in. centers, buried 4 in. in the concrete and laid in a continuous coil. Heat is supplied by an oil-fired hot water boiler.

This installation has six zones of floor panel, four being in the manufacturing space and two in the office section. In addition to the two floor panel zones, the office section includes some wall panels.

The control systems furnished for the 1951-1952 heating season provided the following methods of control:

System Number 1: Boiler water temperature varying with outdoor temperature, with proportioning control on three-way mixing valve for each zone.

System Number 2: Constant boiler water temperature, with proportioning control on three-way mixing valve for each zone.

System Number 3: Boiler water temperature varying with outdoor temperature, three-way mixing valves operating as on-off valves. Fig. 4 shows both the panel piping layout and the control piping diagram for the manufacturing section only.

TABLE 1—CONTROL METHODS FOR INSTALLATION *C*

MODE OF OPERATION			CONTROL MODE SELECTORS POSITION	ZONE SELECTOR POSITION	THERM SELECTOR POSITION	CONTROLLED DEVICE SELECTOR POSITION
CONTROLLED DEVICE	THERMOSTAT TYPE	WATER TEMP				
Motorized Valve on each of 4 zones with continuous circulator operation	Electronic	Fixed	1	—	Electronic	Motorized Valve
	Electronic	Reset	2	—	Electronic	Motorized Valve
	Electric	Fixed	1	—	Electric	Motorized Valve
	Electric	Reset	2	—	Electric	Motorized Valve
Circulator controlled from any one of 4 zones or hall electric thermostat	Electronic	Fixed	3	Desired Zone	Electronic	Burner-Circulator
	Electronic	Reset	4	Desired Zone	Electronic	Burner-Circulator
	Electric	Fixed	3	Desired Zone	Electric	Burner-Circulator
	Electric	Reset	4	Desired Zone	Electric	Burner-Circulator
Burner controlled from any one of 4 zones or hall electric thermostat Continuous circulator operation	Electronic	—	5	Desired Zone	Electronic	Burner-Circulator
	Electric	—	5	Desired Zone	Electric	Burner-Circulator
Burner and circulator controlled simultaneously from any one of 4 zones or hall electric thermostat	Electronic	—	6	Desired Zone	Electronic	Burner-Circulator
	Electric	—	6	Desired Zone	Electric	Burner-Circulator
Circulator controlled by any one of five electric thermostats	Electric	Fixed	7	Desired Zone	Electric	Burner-Circulator

TYPE *C* INSTALLATION

The installation under study for Type *C* construction is a residence located at Minneapolis. This residence is of wood frame construction, single story with a floor area of approximately 1450 sq ft. The floor slab is divided into two sections of approximately equal areas. One section is over an unexcavated area and is constructed of 5-½ in. of reinforced concrete laid on a 4-in. gravel fill. Perimeter insulation was used and consisted of 1-in. thick edge insulation installed vertically to a depth of 8 in. below top of slab with 2-in. thick load bearing insulation around the outside wall laid on the gravel fill and extending a distance of 3 ft inward from the outside wall.

The other section is over an excavated area and is constructed of 10-in. bar joists, and 20-in. centers, 1-½ in. of low density concrete on expanded wire mesh and 3-½ in. of concrete.

The floor panel heating system consists of ¾-in. copper tubing spaced 12-in. on centers buried 3-½ in. in the concrete. The panel is divided into five circuits and each circuit consists of a continuous coil. Heat is supplied by a gas-fired hot water boiler.

The area over the unexcavated portion includes the living room and bedroom, while the area over the excavated portion includes the bathroom, den, hall, kitchen and utility room.

The residence is divided into four zones which are described as follows:

Zone I: The bathroom has a north exposure with a glass area of approximately 20 percent of the wall area. The floor covering is tile laid directly on the concrete slab.

The den has two exposed walls; the north wall is fully exposed and the west wall is adjacent to an unheated garage. Glass area for the north wall is approximately 25 percent. The floor covering consists of a 40 oz combination rubber and fiber padding and heavy carpeting.

Zone II: The master bedroom has both north and east exposures. Glass area for the north wall is 10 percent of the wall area, and for the east wall is 25 percent. The floor covering is 13/16-in. thick oak parquet flooring laid directly on the concrete.

Zone III: The living room, dinette and hallway have both south and east exposures. Glass area for the east wall is approximately 40 percent of the wall area, and for the south wall approximately 30 percent. The floor covering consists of 40 oz combination rubber and fiber padding and heavy carpeting.

Zone IV: Zone IV consists of the kitchen, breakfast nook and utility room. The kitchen and breakfast nook have both south and west exposures. Glass area for the south wall is 10 percent of the wall area and for the west wall 25 percent. The utility room has partially west exposure with glass area 25 percent of the wall area. The panel piping layout for the residence is shown in Fig. 5. The control systems furnished for the 1951-1952 heating season provided the control methods outlined in Table 1. The wiring diagram for these control systems is shown in Fig. 6.

The piping system for this installation was so designed as to permit the use of controls from any source. This installation also lends itself to the study of the effect of carpeting, wood floor, tile and linoleum floor on panel heat transmission rates.

INSTRUMENTATION

The instrumentation of these projects consisted of the location of 12 thermocouples for installations *A* and *B* with 24 thermocouples for installation *C*. These thermocouples consisted of 24-gage, copper-constantan thermocouple wires, soldered at the hot junction and taped to the surface of the copper pipes. It was felt that due to the high conductivity of copper, the surface reading of the copper pipes would give essentially the same reading as a thermocouple immersed in the heating medium. In order to eliminate the possibility of lead conduction, the thermocouple leads from the hot junction were attached to the surface under measurement for a distance of 36 to 40 in. All thermocouple leads were returned to a central location and connected to 12 point, electronic strip-chart recorders. Those thermocouple leads which were embedded in the ground or concrete slab were encased in a vapor-protective membrane.

The strip-chart recorders were of the automatic type and were serviced and calibrated periodically during the period of the tests. In addition, each point on the recorder printed every 30 sec, giving a complete record every 6 min. The calibrated accuracy of the recorder is $+1/5$ of 1 percent of the scale span; however, physical limitations of the chart size permit a reading accuracy of $\frac{1}{2}$ deg.

The instrumentation of both installations *A* and *B* included the following thermocouple locations for continuous temperature recordings:

1. Outside dry bulb temperature
2. Recorder case temperature

Space Temperatures (Located Vertically in Line)

3. Dry bulb temperature at thermostat
4. Dry bulb temperature at 60-in. level
5. Dry bulb temperature at 30-in. level

Panel Surface Temperatures

6. Over outlet pipes
7. Between outlet pipes
8. Over inlet pipes
9. Between inlet pipes

Water Temperatures (At Pipe Surfaces)

10. Outlet from panel
11. Inlet to panel
12. At boiler

The instrumentation for installation *C* included the following thermocouple locations for continuous temperature recordings for each zone:

Water Temperatures (At Pipe Surfaces)

1. Panel supply water temperature for each zone
2. Return water temperature for each zone
3. Boiler water supply
4. Boiler water return

Other Temperatures and Readings Recorded

5. Room dry bulb temperature at thermostat in each zone
6. Outside dry bulb
7. Heat meter readings
8. Temperatures at the top and bottom of slab, and under insulation at each of the heat meter locations.

For the purpose of future study, low-resistance, low-inertia, heat flow meters of the type developed at the A.S.H.V.E. Research Laboratory[1] and thermocouples were installed during construction. The locations of the heat flow meters are as follows:

1. In center section of *Zone II*, at the top of concrete slab and at the bottom of the concrete slab.
2. Along outside wall of *Zone III*, at top of concrete slab, at bottom of concrete slab, and under the 2-in. thick perimeter insulation.
3. In center section of *Zone IV*, at top of concrete slab, and on basement ceiling.

TESTING PROCEDURE

During the 1951-1952 heating season, an average time period of approximately two weeks was chosen for the operation of each control system in the three installations.

The strip-charts were periodically removed from the recorders and reviewed at the University of Minnesota. The equipment employed in reviewing the charts consisted of a special table which facilitated the inspection of the charts by mechanically unrolling them across an illuminated ground glass surface.

From the chart covering the period of operation of a specific control system, several periods were chosen to represent various temperature ranges of outside temperature. In general, for these temperature ranges the most severe temperature fluctuations were selected, since it was felt that these fluctuations imposed the most difficult conditions upon the control system.

RESULTS

Due to the large number of temperatures recorded on the strip-chart, it was difficult to present this information in chart form which would be of practical value; therefore, in reviewing the charts an attempt was made to pick out those points which represented the trends of temperatures shown on the strip-chart. These points, plotted on the curves in Figs. 7 to 16, indicate major deviations from equilibrium conditions. For convenience they have been connected by straight lines. The curves, when compared with the original recorder charts, clearly indicate the results obtained without plotting each point.

The test data recorded are identified by a numbered circle at the end of each curve and may be checked against the code which appears on the right hand side of the chart. In the charts for installation *A* the number 5 which is noted as boiler water temperature on the code is actually converter water temperature.

[1] A Low-Inertia, Low-Resistance Heat Flow Meter, by R. G. Huebscher, L. F. Schutrum, and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 275).

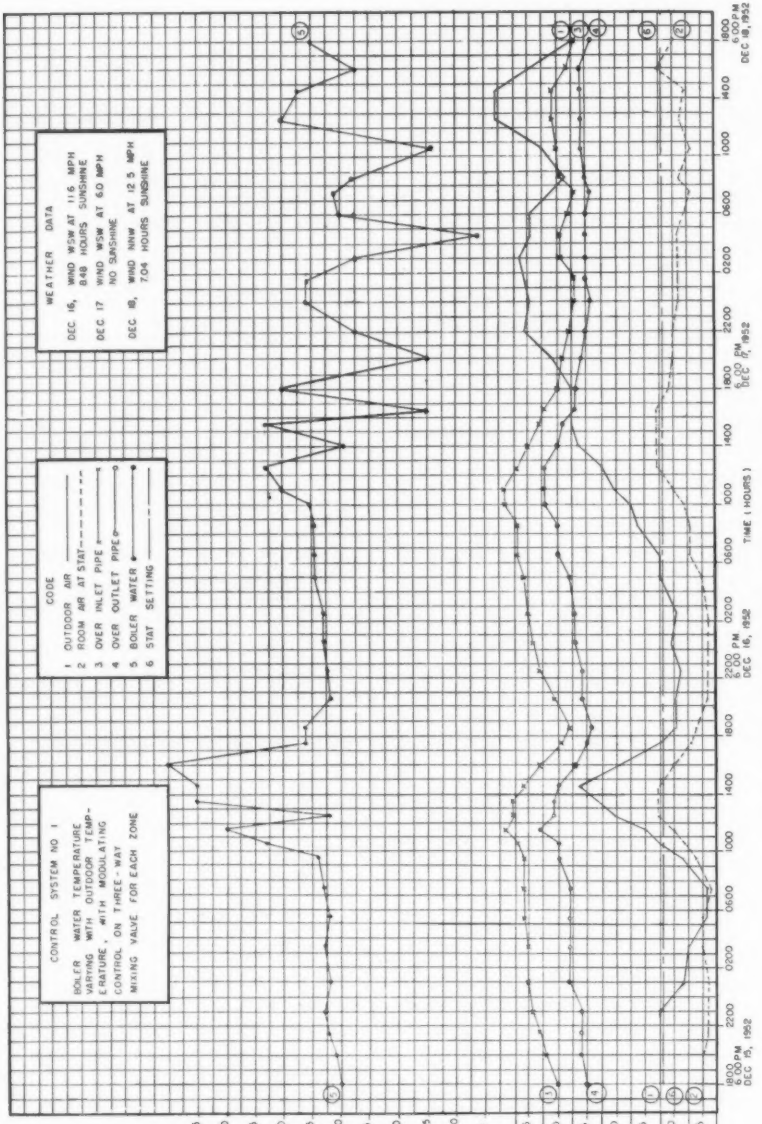

CONTROL SYSTEM NO. 1
BOILER WATER TEMPERATURE VARYING WITH OUTDOOR TEMPERATURE, WITH MODULATING CONTROL ON THREE-WAY MIXING VALVE FOR EACH ZONE
CODE
1 OUTDOOR AIR
2 ROOM AIR AT STAT
3 OVER INLET PIPE
4 OVER OUTLET PIPE
5 BOILER WATER
6 STAT SETTING
WEATHER DATA
DEC 16, WIND WSW AT 11.6 MPH
8.48 HOURS SUNSHINE
DEC 17 WIND WSW AT 6.0 MPH
NO SUNSHINE
DEC 18, WIND NNW AT 12.5 MPH
7.04 HOURS SUNSHINE
TIME (HOURS)
1800 6:00 PM DEC 15, 1952
2200 10:00 PM DEC 15, 1952
0200 2:00 AM DEC 16, 1952
0600 6:00 AM DEC 16, 1952
1000 10:00 AM DEC 16, 1952
1400 2:00 PM DEC 16, 1952
1800 6:00 PM DEC 16, 1952
2200 10:00 PM DEC 16, 1952
0200 2:00 AM DEC 17, 1952
0600 6:00 AM DEC 17, 1952
1000 10:00 AM DEC 17, 1952
1400 2:00 PM DEC 17, 1952
1800 6:00 PM DEC 17, 1952
2200 10:00 PM DEC 17, 1952
0200 2:00 AM DEC 18, 1952
0600 6:00 AM DEC 18, 1952
1000 10:00 AM DEC 18, 1952
1400 2:00 PM DEC 18, 1952
1800 6:00 PM DEC 18, 1952
165 160 155 150 145 140 135 130 125 120 115
95 90 85 80 75 70 65
30 25 20 15 10 5 0 -5 -10

FIG. 7. TEMPERATURE TRENDS FOR INSTALLATION B

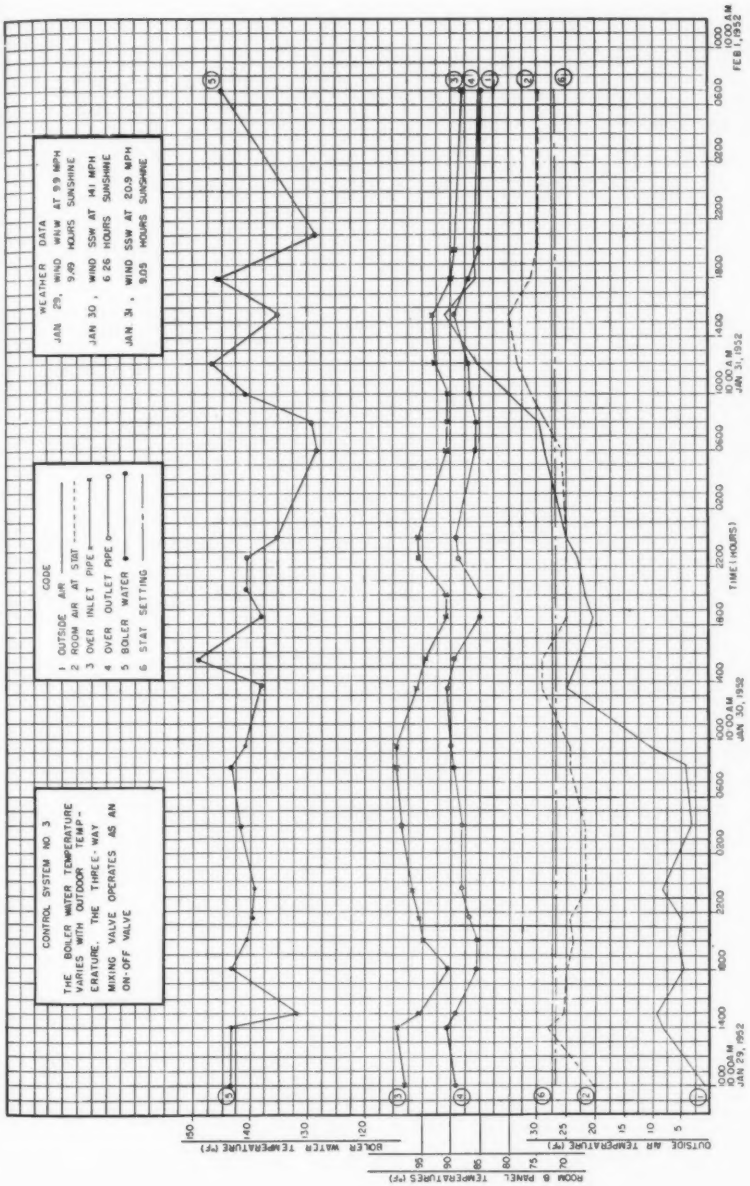

CONTROL SYSTEM NO 3
THE BOILER WATER TEMPERATURE VARIES WITH OUTDOOR TEMPERATURE. THE THREE-WAY MIXING VALVE OPERATES AS AN ON-OFF VALVE
CODE
1 OUTSIDE AIR
2 ROOM AIR AT STAT
3 OVER INLET PIPE
4 OVER OUTLET PIPE
5 BOILER WATER
6 STAT SETTING
WEATHER DATA
JAN 29, WIND WNW AT 9.9 MPH
9.49 HOURS SUNSHINE
JAN 30, WIND SSW AT 14.1 MPH
6.26 HOURS SUNSHINE
JAN 31, WIND SSW AT 20.9 MPH
9.05 HOURS SUNSHINE
BOILER WATER TEMPERATURE (°F)
ROOM & PANEL TEMPERATURES (°F)
OUTSIDE AIR TEMPERATURE (°F)
TIME (HOURS)
1000 10:00 AM JAN 29, 1932
1000 10:00 AM JAN 30, 1932
1000 10:00 AM JAN 31, 1932
1000 10:00 AM FEB 1, 1932

FIG. 8. TEMPERATURE TRENDS FOR INSTALLATION B

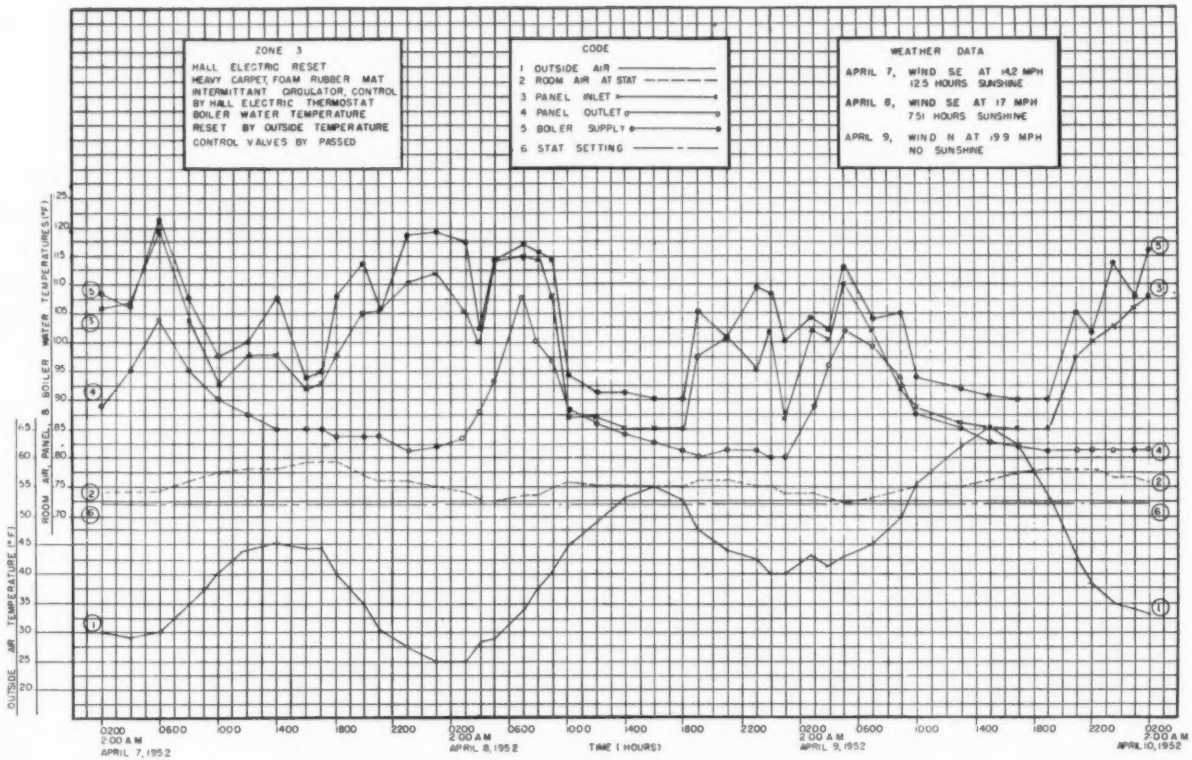


FIG. 9. TEMPERATURE TRENDS FOR INSTALLATION C

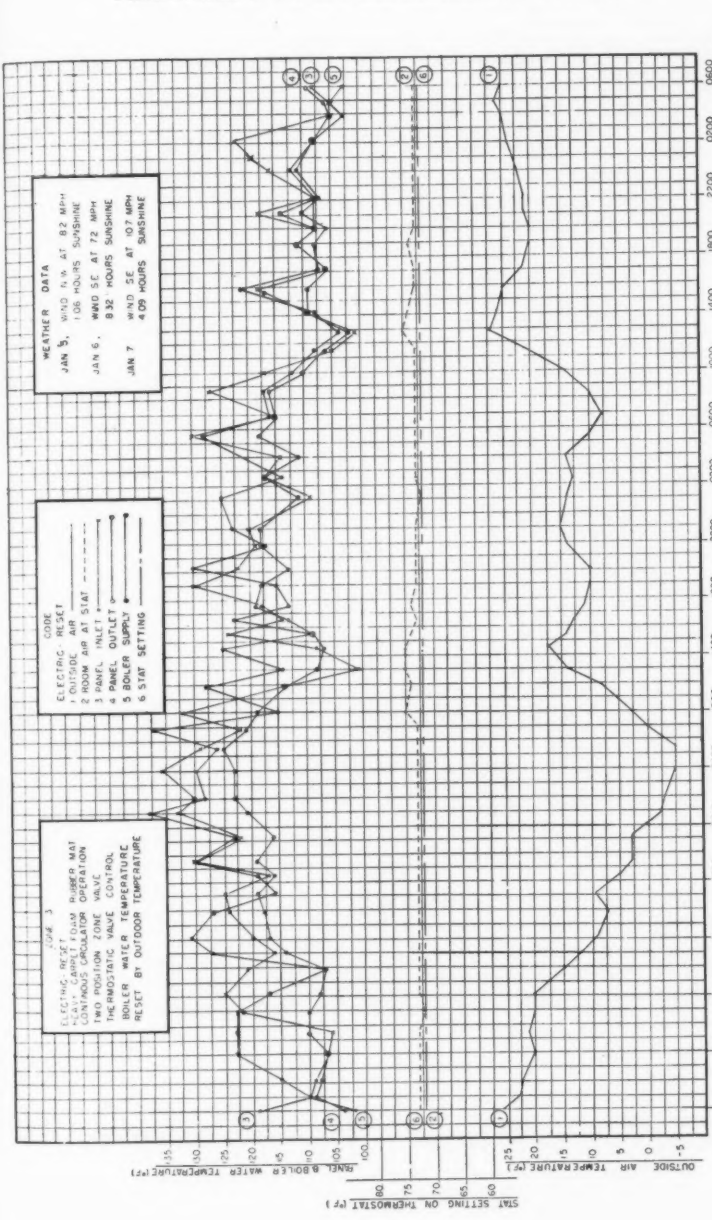

ZONE 3
ELECTRIC RESET
PLAIN CARPET FOAM RUBBER MAT
FLOOR CONSTRUCTION
TWO POSITION ZONE VALVE
THERMOSTATIC VALVE CONTROL
BOILER WATER TEMPERATURE
RESET BY OUTDOOR TEMPERATURE
CODE
ELECTRIC RESET
1 OUTSIDE AIR
2 ROOM AIR STAT
3 PANEL INLET
4 PANEL OUTLET
5 BOILER SUPPLY
6 STAT SETTING
WEATHER DATA
JAN 5, WIND N N W AT 82 MPH
106 HOURS SUNSHINE
JAN 6, WIND SE AT 72 MPH
832 HOURS SUNSHINE
JAN 7 WIND SE AT 107 MPH
409 HOURS SUNSHINE
OUTSIDE AIR TEMPERATURE (°F)
STAT SETTING ON THERMOSTAT (°F)
PANEL & BOILER WATER TEMPERATURE (°F)
TIME (HOURS)
0600 6:00 A.M. JAN 5, 1952
0600 6:00 A.M. JAN 6, 1952
0600 6:00 A.M. JAN 7, 1952
0600 6:00 A.M. JAN 8, 1952

FIG. 10. TEMPERATURE TRENDS FOR INSTALLATION C

It would be well to note that number 6 represents the setting of the space thermostat and not the control point which may vary according to calibration or degree of load.

The operation of control system number 1 for installation *B* is illustrated in Fig. 7. For the period 6:00 p.m., December 15 to 7:30 a.m., December 16, the space temperature was well below the control point setting as a result of the low outside air temperature. The reset temperature of the boiler water, as determined by the ratio set on the indoor-outdoor controller, was not attained because of the excessive demand on the boiler; however, with the rise of the outside temperature to 14 F at 2:30 p.m., December 16, the demand allowed the boiler water to attain the reset temperature. A similar condition in which the boiler is unable to supply the demand exists immediately following the drop in the outdoor air temperature for the period from 5:30 p.m., December 16 to 10:00 a.m., December 17. For the remaining period covered, the rise in outside air temperature permits the boiler water to reach its reset value and supply heat as needed. This condition is indicated by the decreased temperature drop across the panel. Due to the preceding cold period, the wall temperatures were lowered. The following rise in outside air temperature lowered the reset temperature of the boiler water. These two conditions account for the lowered space temperature during the period from 2:30 p.m., December 17 to 2:30 p.m., December 18 because the cold walls absorbed more heat from the room air than was being supplied by the panel.

The operation of control system number 3 for the same installation is shown in Fig. 8. This shows the same condition as shown in Fig. 7, concerning the ability of the boiler to maintain the boiler water temperature at the reset value as a function of the load.

In addition, the outdoor thermocouple for this installation is located on the west wall, and thus does not sense the solar effects until almost mid-day. The control system is consequently unable to compensate for the solar effects in sufficient time to prevent the space from being overheated. This effect is illustrated in the period covered by Fig. 8 and can be seen by noting the space temperatures between the hours of 10:00 a.m. to 3:30 p.m. on January 31.

During the period of 4:00 p.m., January 31 to 7:00 a.m., February 1, the space temperature continues to exceed the thermostat setting, since the recirculated water from all zones to a common return is at a greater temperature than is required by this zone.

Fig. 9 illustrates the intermittent operation of the circulator for installation *C* during a period in which the outside temperature is continually changing. For comparison, Fig. 10 illustrates the continuous operation of the circulator under a similar load condition. Observation of Figs. 9 and 10 shows: *First,* that the temperature drop across the panel is less with continuous circulator operation; *Second,* that a more stable room air temperature exists with continuous circulator operation.

Since stability of the room dry bulb temperature can also be considered as the criterion of control, the room air temperatures of all four zones in installation *C* were plotted for the control system utilizing both reset and fixed boiler water temperature. These two control systems are presented for comparison in Fig. 11 and Fig. 12. Fig. 12 illustrates the control system utilizing the fixed boiler water temperature and reveals the fact that as the outside temperature

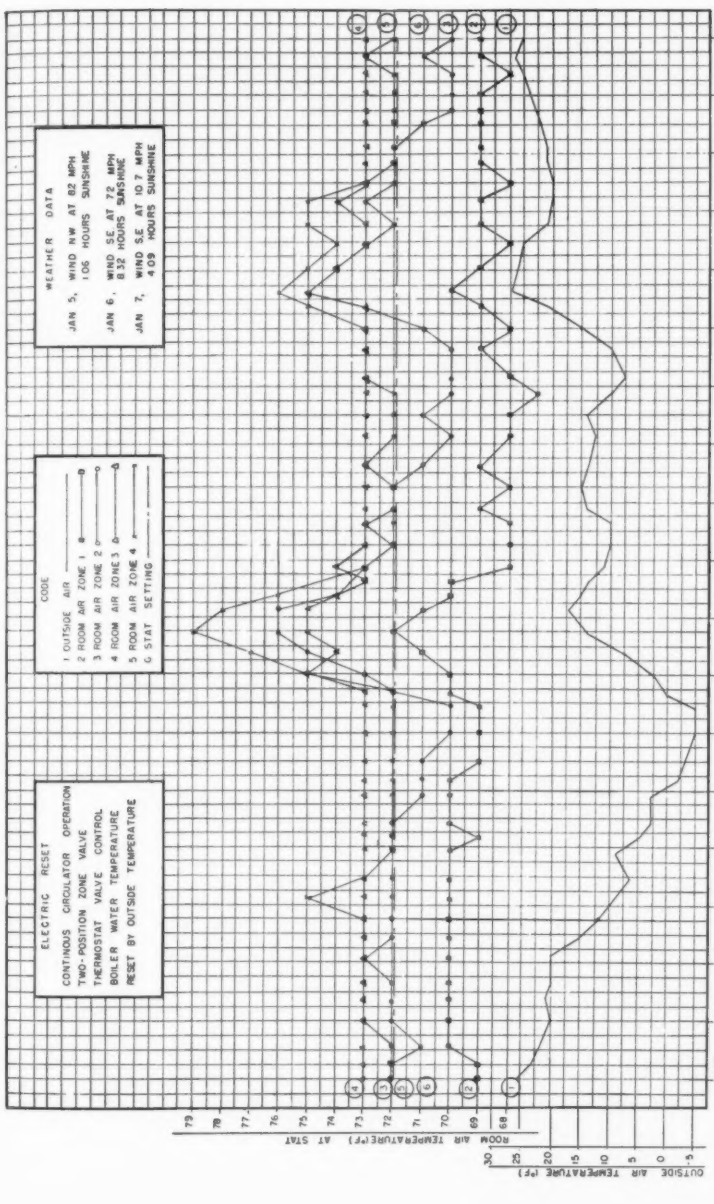


FIG. 11. COMPARISON OF ZONE ROOM TEMPERATURES FOR INSTALLATION C

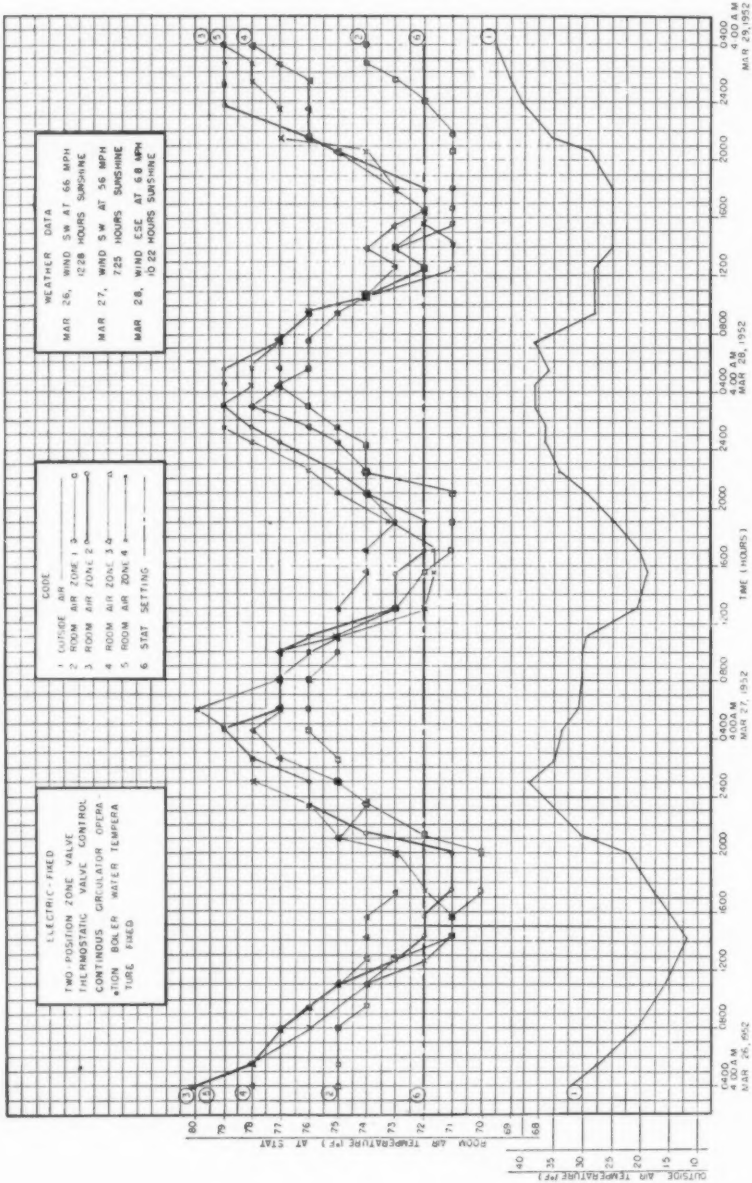

ELECTRIC-FIRED
TWO-POSITION ZONE VALVE
THERMOSTATIC VALVE CONTROL
CONTINUOUS CIRCULATOR OPERA-
TION BOILER WATER TEMPERA-
TURE FIXED
CODE
1 OUTSIDE AIR
2 ROOM AIR ZONE 1
3 ROOM AIR ZONE 2
4 ROOM AIR ZONE 3
5 ROOM AIR ZONE 4
6 STAT SETTING
WEATHER DATA
MAR 26, WIND SW AT 66 MPH
1228 HOURS SUNSHINE
MAR 27, WIND SW AT 56 MPH
725 HOURS SUNSHINE
MAR 28, WIND ESE AT 68 MPH
10 22 HOURS SUNSHINE
ROOM AIR TEMPERATURE (°F.) AT STAT
OUTSIDE AIR TEMPERATURE (°F.)
TIME (HOURS)
0400 4 00 A M MAR 26, 1952
0400 4 00 A M MAR 27, 1952
0400 4 00 A M MAR 28, 1952
0400 4 00 A M MAR 29, 1952

FIG. 12. COMPARISON OF ZONE ROOM TEMPERATURES FOR INSTALLATION C

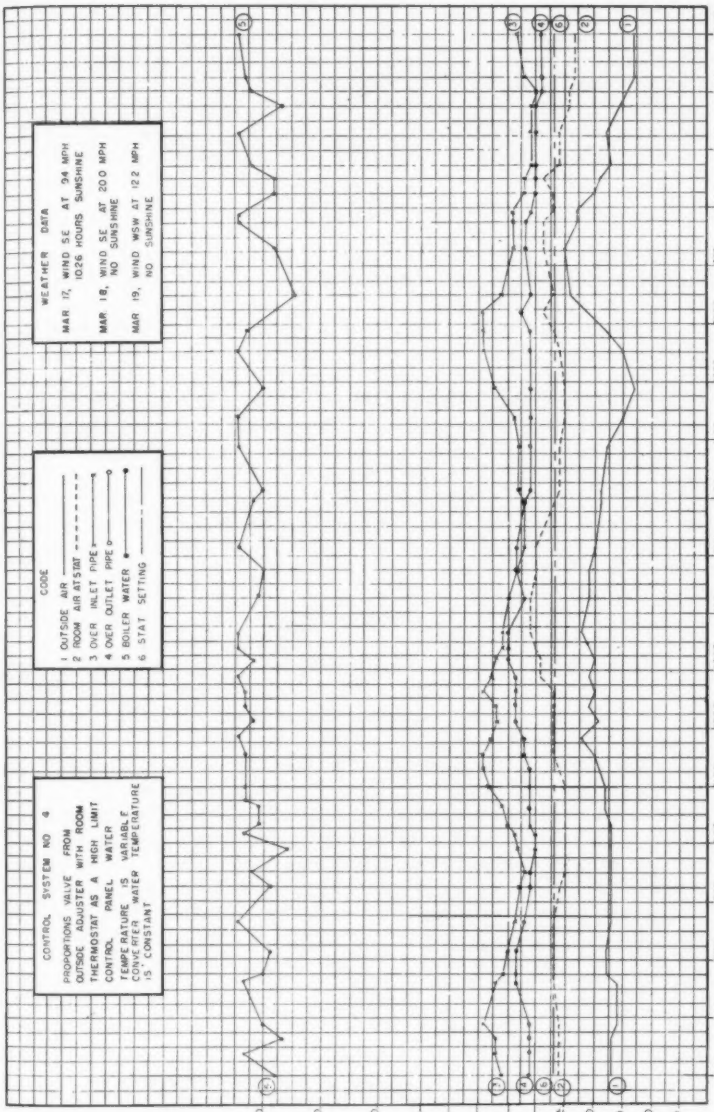

CONTROL SYSTEM NO 4
PROPORTIONS VALVE FROM OUTSIDE ADJUSTER WITH ROOM THERMOSTAT AS A HIGH LIMIT CONTROL PANEL WATER TEMPERATURE IS VARIABLE CONVERTER WATER IS CONSTANT
CODE
1 OUTSIDE AIR
2 ROOM AIR STAT
3 OVER INLET PIPE
4 OVER OUTLET PIPE
5 BOILER WATER
6 STAT SETTING
WEATHER DATA
MAR 17, WIND SE AT 9.4 MPH 1026 HOURS SUNSHINE
MAR 18, WIND SE AT 20.0 MPH NO SUNSHINE
MAR 19, WIND WSW AT 12.2 MPH NO SUNSHINE
BOILER WATER TEMPERATURES (°F)
ROOM & PANEL TEMPERATURES (°F)
OUTSIDE AIR TEMPERATURE (°F)
TIME (HOURS)
MAR 17, 1952
MAR 18, 1952
MAR 19, 1952
MAR 20, 1952

FIG. 13. TEMPERATURE TRENDS FOR INSTALLATION A

fluctuates, wide variation in the space temperature results. Figs. 13 and 14 illustrate performance of installation *A* under similar control conditions.

Figs. 15 and 16 are presented as a comparison of proportioning and on-off valve operation for installation *A* which is classified as a heavy panel, heavy building construction. Examination of the two charts shows very little fluctuation in panel surface temperature, although Fig. 16 reveals considerable fluctuation in the panel inlet water temperature. This illustrates the damping effect of the heavy concrete slab with the coil buried deeper than normal.

CONCLUSIONS

The data gathered during the last two years, although not complete, have led to certain definite conclusions. The foremost of these is the importance of the proper design and installation of the basic heating system. An illustration of this point is the fact that in installation *B* the returns for all six zones were gathered together and mixed in a common return before any recirculation through the three-way mixing valves. This meant that frequently the temperature of the recirculated water was higher for the given zone than the heat loss from that zone warranted. The adverse effects of this condition are shown in Fig. 8 during the hours of 4:00 p.m., January 31 to 8:00 a.m., February 1. It may be noted that even though the space temperature is considerably above the control setting, inlet panel surface temperature and the outlet panel surface temperature are both well into the heating range and a temperature drop is still shown across the panel. This condition could have been eliminated by complete segregation of the return from each zone.

Another phase of basic panel design is further illustrated in installation *B*, where it may be noted in Fig. 7 that the space temperature is below the control setting except where outside air temperatures permit the boiler to supply sufficient heat. One of the major factors which determines the ability of the heating system to maintain a desirable temperature level within a space is the selection of a boiler with sufficient capacity and recovery rate to satisfy the heat demand under the most severe conditions.

Another consideration in the successful operation of any control system is the selection and location of the various control elements. One of the control elements, the location of which is of vital concern, is the outdoor thermostat. The most effective location of this element cannot be categorically stated for any installation until consideration has been given to the type of construction and local climatic conditions. The outdoor thermostat should be so located as to sense the same prevailing climatic conditions as are affecting the space being controlled. For any installation whose exposed surfaces have a high percentage of glass area, the location of the outdoor thermostat for sensing the solar effects is especially important. Installation *B* illustrates the result of the inability of the outdoor thermostat to sense the solar effects. This is shown in Fig. 8. During the daylight hours of January 31, in which there were only 9.05 hr of sunshine (100 percent of that possible), the early solar effects were not sensed by the outdoor thermostat in sufficient time to prevent overheating of the space.

One means of achieving stable space temperature in buildings classified as being of light construction is the continuous operation of the circulator, regardless of outside air temperature fluctuation. Intermittent circulation with its

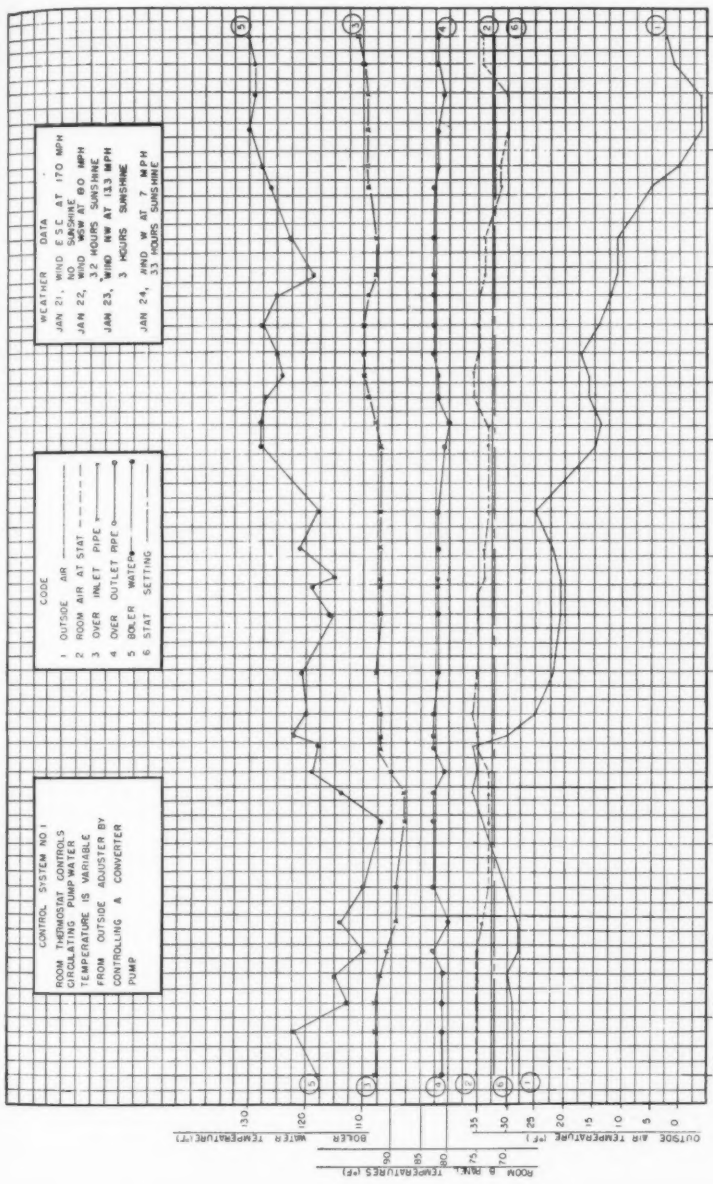


FIG. 14. TEMPERATURE TRENDS FOR INSTALLATION A

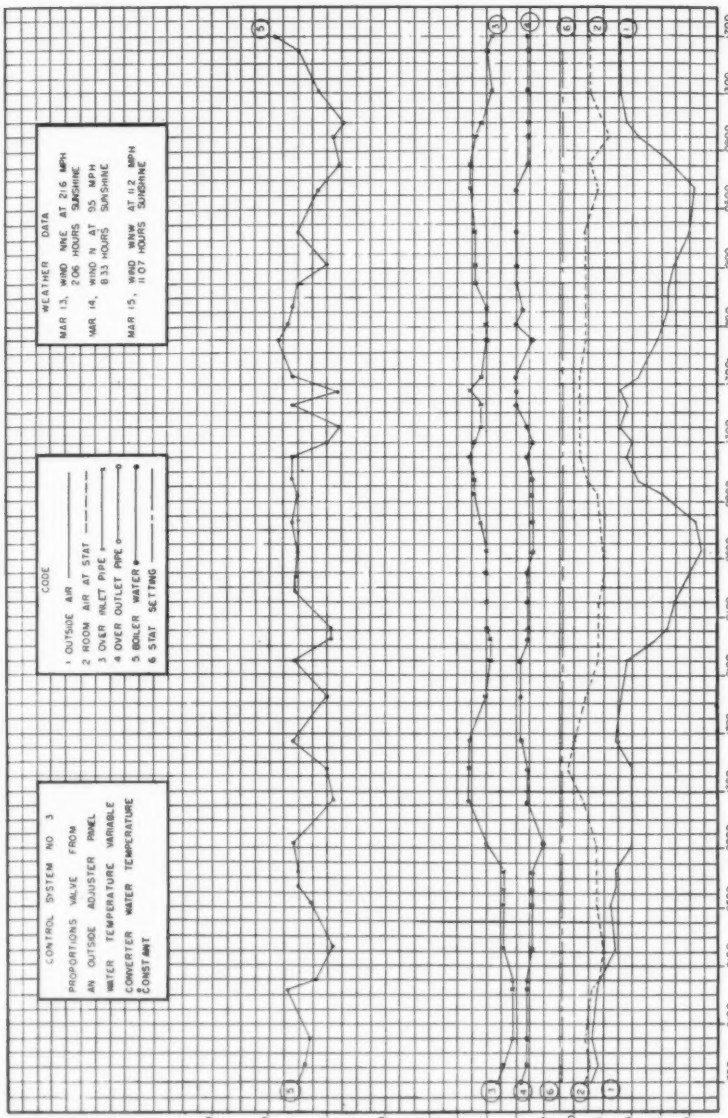


FIG. 15. TEMPERATURE TRENDS FOR INSTALLATION A

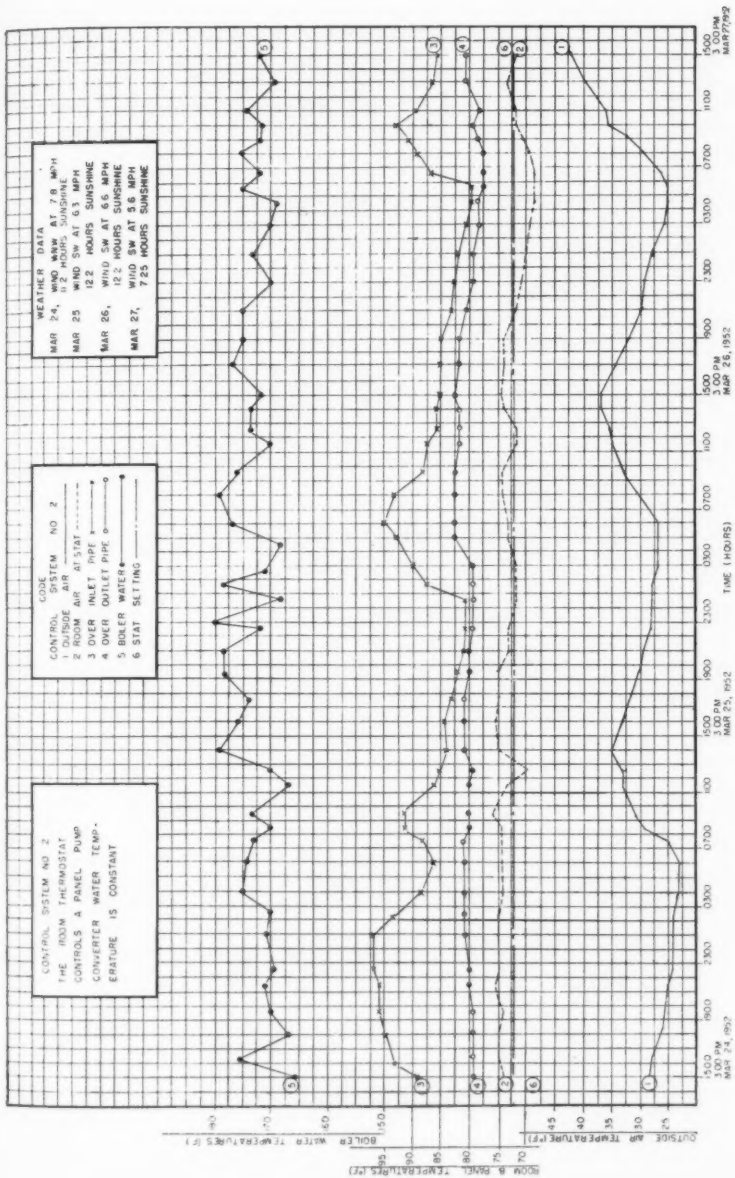

CONTROL SYSTEM NO 2
THE ROOM THERMOSTAT CONTROLS A PANEL PUMP CONVERTER WATER TEMPERATURE IS CONSTANT
CONTROL SYSTEM NO 2
1 OUTSIDE AIR
2 OUTSIDE AIR STAT
3 OVER INLET PIPE
4 OVER OUTLET PIPE
5 BOILER WATER
6 STAT SETTING
WEATHER DATA
MAR 24, WIND WNW AT 7.8 MPH 12.2 HOURS SUNSHINE
MAR 25 WIND SW AT 6.3 MPH 12.2 HOURS SUNSHINE
MAR 26, WIND SW AT 6.6 MPH 12.2 HOURS SUNSHINE
MAR 27, WIND SW AT 5.6 MPH 7.25 HOURS SUNSHINE
ROOM & PANEL TEMPERATURES (F)
BOILER WATER TEMPERATURES (F)
OUTSIDE AIR TEMPERATURE (F)
TIME (HOURS)
MAR 24, 1952
MAR 25, 1952
MAR 26, 1952
MAR 27, 1952

FIG. 16. TEMPERATURE TRENDS FOR INSTALLATION A

associated hot and cold surges of water causes large panel surface temperature cycles which in turn affect the space temperature. The performance of a control system utilizing both continuous and intermittent circulation in this type of installation is exemplified in Fig. 9 and Fig. 10 for installation *C*. These figures indicate that, for operation under the same general load conditions, a control system utilizing the continuous circulation affected more stable space temperature control.

In a control system which employs reset action, the boiler water temperature is adjusted upward or downward a predetermined amount for each degree drop or rise in the outside air temperature. This type of control system permits more desirable stable control of the heat supplied to the space than would a control system using a constant boiler water temperature setting. This fact is illustrated in Fig. 11 and Fig. 12 for the operation of the two control systems in installation *C* under the same general load conditions. A further illustration of this fact as brought out in the operation of the control system for installation *A* is shown in Fig. 13 and Fig. 14.

Installation *A* was the only one of the three installations which utilized both on-off and modulating valve action. It is noted by reviewing Fig. 15 and Fig. 16 that for similar load condition there is relatively no difference in the stability of the space temperature. This fact can possibly be accounted for by the large mass of the concrete panel.

In conclusion it may be said that the continuation of this field test program will provide additional, definite conclusions on the application of automatic temperature controls to floor panel heating systems.

ACKNOWLEDGMENT

The authors wish to thank the Barber-Colman Co., Rockford, Ill., the Johnson Service Co., Milwaukee, and the Minneapolis-Honeywell Regulator Co., Minneapolis, for the controls which they provided and the assistance and counsel they rendered. The authors also thank J. R. Rush, Research Assistant and R. R. Head, Research Fellow at the University of Minnesota, who rendered valuable service during this project.

1473

No. 1473

HEAT EXCHANGES IN A CEILING PANEL HEATED ROOM

By L. F. SCHUTRUM*, G. V. PARMELEE**, AND C. M. HUMPHREYS†,
CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THIS PAPER, first of a series dealing with problems of heat exchanges within panel heated and cooled spaces is restricted to ceiling panel heated spaces in which the entire ceiling area is heated. Most of the tests were made with a simple room arrangement in which all parts of the floor and walls had the same surface temperature. Some tests were made, however, to explore effects of a departure from uniform environment. Results from these are included.

Research concerning heat exchanges within panel heated spaces is part of the long-range program of the A.S.H.V.E. Technical Advisory Committee on Panel Heating and Cooling to develop reliable design data. Work described in this paper had the advisory guidance of the Committee's Group B.‡ Previous papers described the results of research on heat flow within concrete and plaster panels. Experiments were made in the newly completed Environment Laboratory of the A.S.H.V.E. Research Laboratory in Cleveland.

TEST APPARATUS

Interior surfaces of the room [1] consisted of 75 aluminum panels of various sizes, arranged to provide maximum flexibility in surface temperature control. All room side surfaces were painted with a semi-gloss gray paint.

Low-inertia, low-resistance heat flow meters [2] covered approximately 10 percent of the inside surface area of the room. Its dimensions were 24 ft 6 in. by 12 ft, with an 8 ft ceiling height.

---

* Research Fellow, A.S.H.V.E. Research Laboratory. Junior member of A.S.H.V.E.

** Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

† Senior Engineer, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

‡ *Personnel:* J. M. van Nieukerken, *Chairman;* C. O. Mackey, *Vice Chairman;* C. M. Ashley, C. F. Boester, W. P. Chapman, W. S. Harris, Linn Helander, John W. James, A. T. Jorn, C. F. Kayan, H. A. Lockhart, W. E. Long, R. L. Maher, R. A. Miller, D. L. Mills, G. L. Wiggs.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

Surface temperatures were measured by thermocouples cemented in grooves cut into the backs of the aluminum panels. Air temperatures, measured by butt-soldered 36-gage copper constantan thermocouples, were taken at six locations along two walls and at various levels in the center and near one end of the room.

Infiltration was recognized as an important factor affecting both room air temperatures and panel outputs. Infiltration air entered the room through 6 vertical 2-in. diameter perforated pipes, closed at the top. It was metered by means of an orifice. Four of the six pipes were located along one long wall (east), and 2 along one short wall (south). To define the exit of this air as completely as possible, the room was fairly tightly sealed and the door cracks taped. About 50 percent of the infiltration air left the room at the floor level through a 6 ft long slot, about 0.4 sq ft in area, in the center of the north half of the west wall. The air temperature was taken at this point. The remaining 50 percent of infiltration air undoubtedly left the room through small cracks between room surface panels. In a normal house or building the actual entrance and exit temperatures of infiltrating air are undetermined.

TEST PROCEDURE

All data in this report were taken under steady state conditions, usually with a uniform environment. Panel temperatures and infiltration rates and temperatures were controlled: room air temperatures and heat flow rates were dependent variables.

In this report, *uniform environment* means that all panels of the four walls and the floor were maintained at the same surface temperature. *Non-uniform environment* indicates that floor and wall surfaces were not at the same temperature. *Neutral surface* is a room surface which was neither heated nor cooled by liquid circulation, but was allowed to seek its own temperature level. The back of each panel being insulated, and temperature of air behind the insulated panel controlled, heat flow under these conditions was very small.

Most of the tests described herein were made with an unoccupied, unlighted test room, containing only necessary instrumentation. A few tests were made with the room lighted and furnished to simulate a normal living room.

CALIBRATION

Heat flow meters were individually calibrated in place with a portable calibrator,[2] which assured precise heat flow meter measurements at each such meter location. It was necessary, however, to investigate the integrating and averaging ability of the meter for the panel area surrounding it. Twenty-four 500-watt incandescent lamps were installed in the room. Electrical energy input and heat outflow from the room were measured by the heat flow meters. The lamps were distributed uniformly over the room, but in one test, all of the heat was supplied to one half of the room. The average difference between metered heat input and output rates in these tests was approximately 5 percent, with the heat flow meters showing the higher value.

The reliability of heat flow measurements in the Environment Laboratory was also proved by the heat balance obtained in each test. Heat flowing into the laboratory from the heated ceiling panel must equal the heat flow out through cooled or neutral panels for conditions of no infiltration and no other internal heat sources. In 26 tests with uniform environment the average difference

between the metered heat input and heat output was less than one percent of the total heat flow. Maximum difference was within ±5 percent; the difference in 10 of the 26 tests was less than one percent. In general, the measured heat flow from the ceiling panels was slightly lower than the total heat flow outward through the other surfaces.

TEST RESULTS—UNIFORM ENVIRONMENT

Heat Flow—No Infiltration: The first series of tests in the Environment Laboratory was made with uniform environment, no infiltration air, the entire

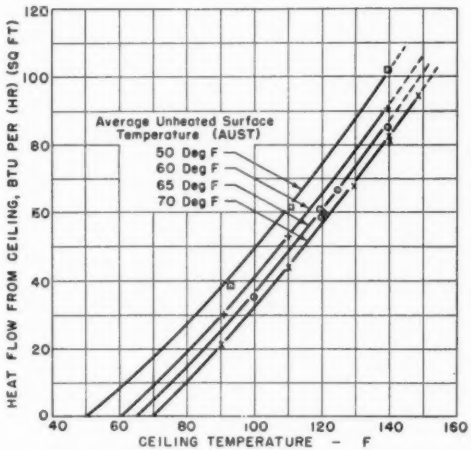


FIG. 1. HEAT FLOW FROM THE CEILING VS. CEILING TEMPERATURE AND AUST

No infiltration—uniform environment (24½ x 12 x 8 ft high room)

ceiling heated, and the room dark and unoccupied. Heat flow rates are shown in Fig. 1 as a function of the ceiling temperature and the area weighted average temperature of the unheated surfaces of the room. This *average unheated surface temperature* will be referred to hereafter as the AUST. Experimental temperatures were not always at the nominal curve temperatures and thus the curves do not pass through all the points. Except for the door temperature, which was not controlled, the individual panel temperatures were within ± one deg of the AUST.

Table 1 shows temperature and heat flow data for Test No. 34. The distribution of heat leaving the room through the cooled surfaces is typical of the tests with a uniform environment and no infiltration. The distribution is given for each surface as total heat flow, percent of ceiling input, and heat flow per square foot of surface. Calculated net radiant energy exchange between each surface and the remaining room surfaces is expressed as a percentage of the net exchange between the ceiling and the entire room. Calculations are based upon the

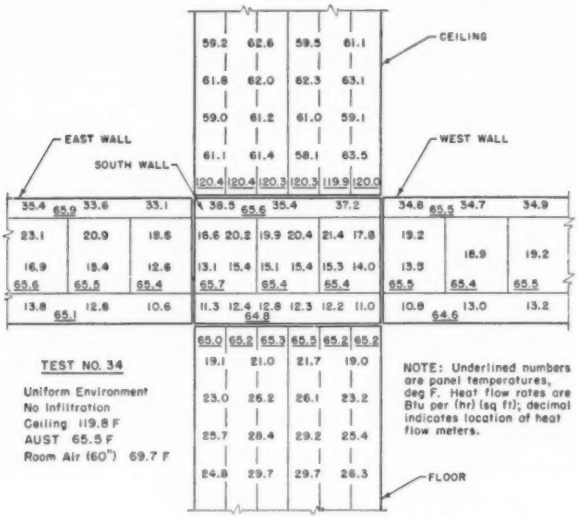

CEILING
EAST WALL
SOUTH WALL
WEST WALL
FLOOR
TEST NO. 34
Uniform Environment
No Infiltration
Ceiling 119.8 F
AUST 65.5 F
Room Air (60") 69.7 F
NOTE: Underlined numbers are panel temperatures, deg F. Heat flow rates are Btu per (hr) (sq ft); decimal indicates location of heat flow meters.

FIG. 2. HEAT FLOW AND TEMPERATURE DISTRIBUTION FOR SOUTH HALF OF ENVIRONMENT LABORATORY

Uniform environment, no infiltration (24½ x 12 x 8 ft high room)

TABLE 1—HEAT FLOW DISTRIBUTION FOR UNIFORM ENVIRONMENT WITH NO INFILTRATION

Test No. 34 (AUST: 65.5 F)

Air Temperatures at Center of Room: 67.3 F at 30-in. level; 69.7 F at 60-in. level

SURFACE	SURFACE TEMP F	TOTAL HEAT FLOW BTU/HR	PERCENTAGE OF CEILING HEAT FLOW	HEAT FLOW BTU/HR (SQ FT)	ANGLE[a] FACTOR	PERCENTAGE[b] RADIANT ENERGY EXCHANGE
Ceiling	119.8	17,995	100.0	59.8		100.0
Floor	65.3	7,585	42.1	25.2	0.425	42.8
North (incl door)	67.3	1,210	6.7	12.7	0.090	8.8
South	65.3	1,755	9.8	18.3	0.090	9.2
West	65.3	3,765	20.9	19.2	0.197	19.3
East	65.2	3,665	20.4	18.7	0.197	19.9
Total:						
Walls and Floor		17,980	99.9			
Unaccounted for		−15	+0.1			

[a] A geometrical factor which takes into account both the shape and the relative position of the surfaces. It represents, as a fraction, the radiant energy emitted from the ceiling which is intercepted by each surface, with no allowance made for reflections. See Reference 4.

[b] This is the calculated net radiant energy exchange between each surface and the rest of the room. Calculation based on emissivity = 1.0.

observed surface temperatures and black body radiation. Since the exact emissivity of the room surfaces is not yet known and since the results of the calculations are expressed as percentages, no further refinement in calculation seemed justified. Angle factors for the surfaces with respect to the ceiling are also included. The percent black body radiation exchange, given in the last column, compares favorably with the percent of ceiling heat flow. This indicates that the heat transfer occurs largely by radiation.

Since no infiltration air was supplied in this test, heat input to the room from the ceiling should equal the heat flowing from the room through the other five surfaces. Table 1 indicates a difference of only 15 Btu per hr in these two values.

Fig. 2 shows surface temperature and heat flow distribution in the south half of the test room, also for Test No. 34. The average surface temperature of each panel and the heat flow rate in Btu per (hr) (sq ft) at each heat flow meter location are noted. This shows that although the average temperatures of the individual panels of the walls and floor are very nearly the same, the heat flow rates are much higher at the top of the walls near the heated ceiling than near the floor. This would probably be different in practice because the wall surface temperature would not be the same at all points.

Heat Flow—With Infiltration: The effect of infiltration on heat flow rates is illustrated by Fig. 3 where the results of tests in which the rate of infiltration was varied while the ceiling temperature was held at 120 F and the other surfaces at 65 F are shown. The temperature of the infiltration air was held at zero F. The ceiling heat flow increases slightly with an increase in infiltration rate; conversely, the other five surfaces show a decrease in heat flow. These heat flows are influenced both by increased air movement and decreased room air temperature. The number of air changes is based upon the density at the temperature of the infiltration air. Table 2 shows the heat flow distribution for Test No. 61, one of the series shown in Fig. 3. Note that although surface temperatures are nominally the same as in Test No. 34, Table 1, the heat flow distribution is considerably altered.

Additional heat flow, Q_c, from the ceiling due to infiltration air is shown by the curves of Fig. 4. For ceiling temperatures and AUST other than 120 F and 65 F, respectively, approximate corrections can be made as noted in the figure.

Heat Transfer to the Air: With no infiltration net heat transfer to the air is zero, that is the heat transferred from the warm surfaces to the air equals the heat transferred from the air to the cold surfaces. With infiltration there is a steady flow of air through the heated space producing a heating load equal to:

1. The difference between total heat inflow from the ceiling and the heat flow out through the walls and floor, or

2. The sum of differences in heat flow for each surface with and without infiltration, if the surface temperatures are identical in each case, or

3. The weight rate of flow times the specific heat at constant pressure times the difference between the temperatures of the exit and the entering air.

Each method should yield identical results. Heat transferred to the air shown in Fig. 3 was substantially the same whether Method 1 or 2 was used. Had all the infiltration air left the room in such a way that its temperature could have been determined, heat transferred to the air could have been determined by Method 3. Since the exit air temperature was known for only the 50 percent

of the air which left the room through the 6-ft slot, Method 3 could not be used without some approximation. If, however, the temperature of the air leaving through the cracks between the panels was assumed to be equal to the area-weighted average temperature of all room surfaces, and this was averaged with the measured exit temperature, Method 3 gave good correlation with the other two methods, and with considerably less experimental scatter. This calculated average exit air temperature was in all cases higher than the air temperature

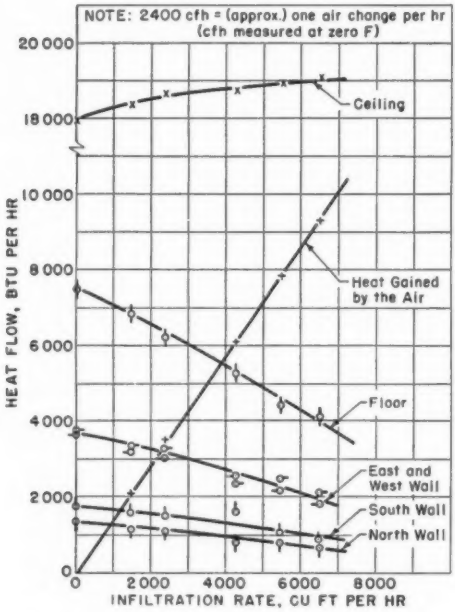


FIG. 3. EFFECT OF INFILTRATION RATE ON HEAT FLOW

Ceiling at 120 F, all other surfaces at 65 F; 0 F infiltration air temperature (24½ x 12 x 8 ft high room)

at the 60 in. level at the center of the room. The curves of Fig. 5 were determined by Method 3 and show that heat gain by the infiltration air is dependent upon the rate of change and the temperature of the infiltration air and the *area-weighted average temperature* of all room surfaces. This temperature is hereinafter referred to as the AST.

Values of Fig. 5 apply only to the test room and the manner in which the infiltration air entered and left the room. There is some experimental evidence to show that the latter has a significant effect on heat transfer to the air.

Room Air Temperature—No Infiltration: Fig. 6 shows the air temperature gradients from floor to ceiling for various test conditions. The ordinate is the height above the floor and the abscissa is a dimensionless ratio of temperature

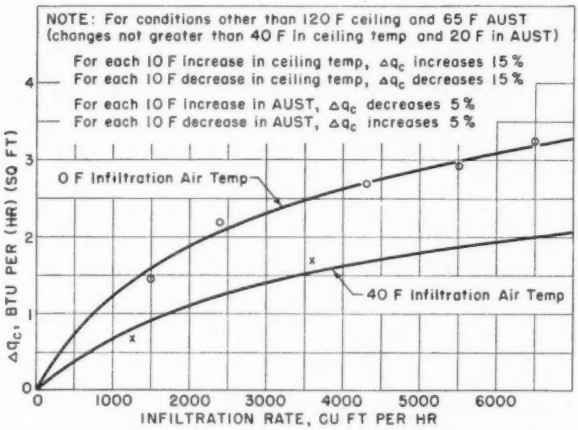

NOTE: For conditions other than 120 F ceiling and 65 F AUST
(changes not greater than 40 F in ceiling temp and 20 F in AUST)
For each 10 F increase in ceiling temp, Δqc increases 15%
For each 10 F decrease in ceiling temp, Δqc decreases 15%
For each 10 F increase in AUST, Δqc decreases 5%
For each 10 F decrease in AUST, Δqc increases 5%
0 F Infiltration Air Temp
40 F Infiltration Air Temp
Δqc, BTU PER (HR) (SQ FT)
INFILTRATION RATE, CU FT PER HR
0
1000
2000
3000
4000
5000
6000
1
2
3
4

FIG. 4. ADDITIONAL HEAT FLOW FROM CEILING DUE TO INFILTRATION AIR

Ceiling at 120 F, all other surfaces at 65 F (24½ x 12 x 8 ft high room)

differences. In tests with a uniform environment and no infiltration the gradient could be represented by Curve A, regardless of ceiling temperature and AUST. On the basis of Curve A, Fig. 7 was constructed to show relationship between air temperature at the 60-in. level and AUST and ceiling temperature. Dew-point temperature had a measurable effect on room air temperature because of absorption of radiation. Air temperatures were therefore corrected empirically

TABLE 2—HEAT FLOW DISTRIBUTION FOR UNIFORM ENVIRONMENT WITH INFILTRATION

Test No. 61—2⅓ Air Changes at Zero F (AUST: 65.5 F)
Air Temperature at Center of Room: 56.0 F at 30-in. level; 61.0 F at 60-in. level. Air Temperature at Exit Slot: 56.2 F

SURFACE	SURFACE TEMP F	TOTAL HEAT FLOW BTU/HR	PERCENTAGE OF CEILING HEAT FLOW	HEAT FLOW BTU/HR (SQ FT)	ANGLE[a] FACTOR	PERCENTAGE[b] RADIANT ENERGY EXCHANGE
Ceiling	119.5	18,955	100.0	63.0		100.0
Floor	65.4	4,450	23.5	14.8	0.425	42.6
North (incl door)	66.4	865	4.5	9.1	0.090	8.9
South	65.3	1,115	5.9	11.6	0.090	9.0
West	65.3	2,490	13.1	12.7	0.197	19.8
East	65.4	2,195	11.6	11.2	0.197	19.7
Heat Gain by Air		7,840[c]	41.4			

[a] A geometrical factor which takes into account both the shape and the relative position of the surfaces. It represents, as a fraction, the radiant energy emitted from the ceiling which is intercepted by each surface, with no allowance made for reflections. See Reference 4.
[b] This is the calculated net radiant energy exchange between each surface and the rest of the room. Calculation based on emissivity = 1.0.
[c] Heat gain by air by difference.

to a 30 F dew-point basis. There can be only one air temperature for a given ceiling temperature and AUST. Fig. 7 shows that to change the air temperature one degree requires a change of 12 deg in ceiling temperature or 1.1 deg change in AUST. The following relationship between AST and Room Air Temperature can be developed from Fig. 6:

$$\text{AST} - \text{Air Temperature} = \left\{ \frac{\text{Ceiling Area}}{\text{Total Surface Area}} - C \right\} (\text{Ceiling Temp.} - \text{AUST})$$

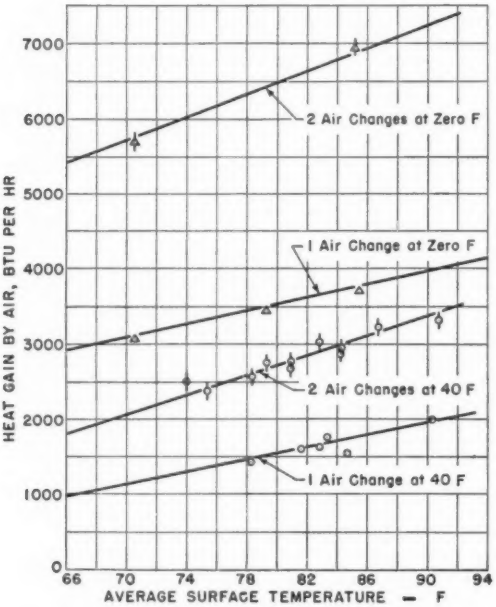


FIG. 5. HEAT GAIN BY INFILTRATION AIR

(24½ x 12 x 8 ft high room)

C is a constant obtained from Curve A of Fig. 6. For the 60 in. level $C = 0.08$.

Tests indicated that the air temperature at the 60 in. level was substantially constant throughout the room. Uniformity of air temperatures at other levels was not investigated.

Room Air Temperature—With Infiltration: Fig. 8 shows the effects of infiltration rate and temperature on room air temperature, and that the room air temperature is lowered with increasing infiltration rate and decreasing infiltration air temperature. This figure presents data for zero and 40 F infiltration air temperatures.

From curves similar to those shown in Fig. 8, an air temperature correction curve, Fig. 9, was obtained. For simplicity, infiltration is given in terms of air

changes instead of the previous volume designation. The ordinate gives the reduction in air temperature which would result from any combination of infiltration rate and temperature. For example, consider a ceiling panel of 118 F and AUST of 67 F. With no infiltration the room air temperature at the 60 in. level would be 71 F (see Fig. 7). If there were one air change per hour at 20 F entering air temperature, the room air temperature would be depressed 3 deg. This would give a room air temperature of 68 F. This correction from Fig. 9 corresponds to a difference of 47 deg between the AUST of 67 F and the infiltration air temperature of 20 F.

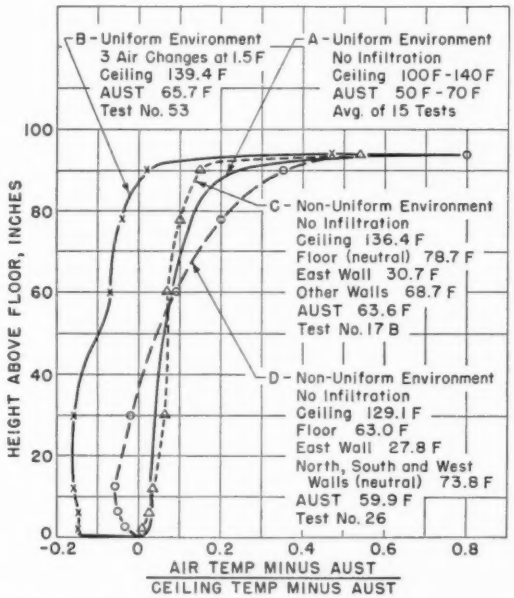


FIG. 6. FLOOR TO CEILING TEMPERATURE DISTRIBUTION AT THE CENTER OF ROOM

(24½ x 12 x 8 ft high room)

No single curve was found which would give the temperature gradient for tests with infiltration. Curve B of Fig. 6 shows the temperature gradient for Test No. 53 with infiltration and uniform environment.

Air Motion: Air velocities were measured by a hot wire anemometer, developed[3] at the A.S.H.V.E. Research Laboratory for investigation of room air distribution. Air velocities in the center of the room at the 3½ in., 30 in., and 60 in. levels were about 10 fpm for the tests without infiltration but increased to about 30 fpm with the higher infiltration rates. Direction of air movement was not determined and since at velocities of the order of 10 fpm, the yaw

correction to the instrument readings may be as much as 25 percent, the values cited are approximate.

No definite air motion existed for the uniform environment without infiltration except for a downward movement within a fraction of an inch of the walls. With infiltration, the entering air tended to drop toward the floor and move in the direction of the exit slot.

Effects of Convection: Heat flow rates shown in Fig. 1 were lower than those commonly thought to prevail in the field. An attempt was made to increase these rates by creating greater air motion. A 15 in. floor fan was placed 60 in.

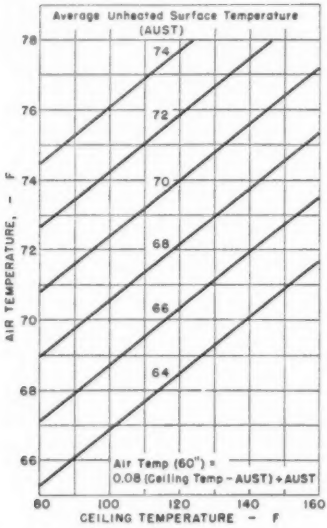


FIG. 7. AIR TEMPERATURE AT 60 IN. LEVEL, CENTER OF ROOM, VS. CEILING TEMPERATURE AND AUST

No infiltration—uniform environment (24½ x 12 x 8 ft high room)

above the floor in the northeast corner of the room and directed toward the center of the ceiling. Table 3 gives a comparison of the heat flow rates obtained with and without the fan in operation. Since surface temperatures for the two tests differed by less than 1 deg, the radiation exchange was essentially the same. The marked increase in total heat flow must, therefore, have resulted from an increase in convection. The particularly large increase of the south wall heat flow was due to the direct impingement of the air stream on this surface.

Emissivity of Room Surfaces: Emissivity of the room surfaces is estimated at about 0.9, although no direct measurements have as yet been made. Radiation exchange between the ceiling and the rest of the room was calculated with this emissivity with some allowance made for reflections. The convective heat trans-

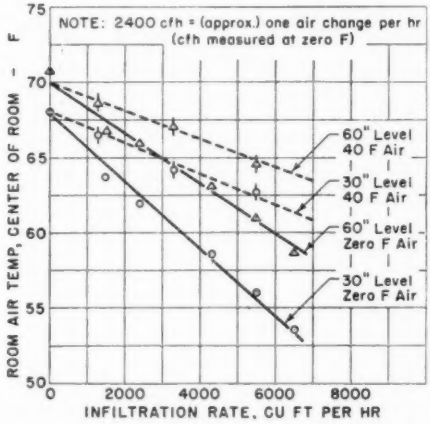


FIG. 8. THE EFFECT OF INFILTRATION AIR RATE AND TEMPERATURE ON ROOM AIR TEMPERATURE

Ceiling 120 F, all other surfaces 65 F (24½ x 12 x 8 ft high room)

fer was then determined as the difference between the total and radiant heat flow. The convection coefficients for the ceiling for these two conditions, based on the temperature difference between the air at the 60 in. level in the room center and the ceiling, are as follows:

with fan h_i = 0.80 Btu per (hr) (sq ft) (F deg).
without fan h_i = 0.04 Btu per (hr) (sq ft) (F deg).

TABLE 3—EFFECT OF AIR MOTION ON HEAT FLOW RATES FOR UNIFORM ENVIRONMENT

SURFACE	AVERAGE HEAT FLOW RATE BTU/(HR) (SQ FT)		PERCENTAGE INCREASE
	FAN NOT OPERATING	FAN OPERATING	
Ceiling	58.9	86.9	47.5
Floor	24.2	32.3	33.5
North Wall	14.0	20.2	44.3
South Wall	19.1	32.3	69.1
East Wall	20.1	29.9	48.8
West Wall	19.5	29.3	50.3
Air Temperature, Fahrenheit, at 60 in. level, center of room			
	71.6	80.7	

Test Conditions—Ceiling temp.: 120 F (nominal)
All other surfaces: 65 F (nominal)
No infiltration

Air velocity at fan:
750 fpm heat input
to fan: 208 Btu/hr

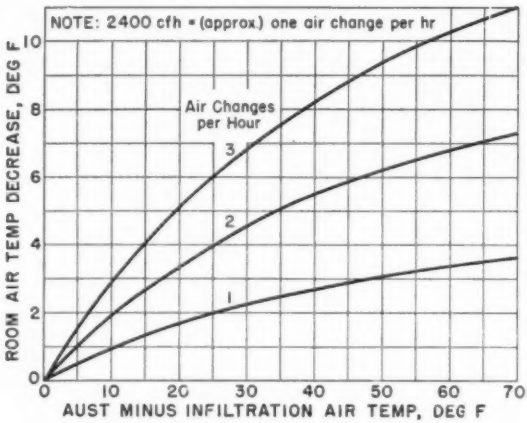


FIG. 9. DECREASE IN ROOM AIR TEMPERATURE AT THE 60 IN. LEVEL VS. INFILTRATION AIR TEMPERATURE AND RATE

Uniform environment (24½ x 12 x 8 ft high room)

Surface Temperature and Wall Conductance: For a uniform environment let a wall conductance, C', be defined as the quotient of the total heat flow through all the unheated surfaces divided by the product of the area of these surfaces times a uniform temperature difference between the inside surface and the outdoor air. Room surfaces, exclusive of the heating surface, can represent an outside wall the conductance of which, for a given inside surface temperature and heat flow rate, varies with the temperature which is assigned to the outside

TABLE 4—THE EFFECT OF NON-UNIFORM ENVIRONMENT ON CEILING HEAT FLOW WITH NO INFILTRATION

TEST SERIES NO.	CONTROLLED VARIABLE	TEST NO.	CEILING	SURFACE TEMPERATURES, F				CEILING HEAT FLOW BTU/(HR) (SQ FT)	
				Floor	Glass (East Wall)	Other Walls	AUST	Non-Uniform	Uniform (Fig. 1)
I	Ceiling Temperature	18	105.9	N[a]65.9	29.4	66.8	58.3	51.5	49.5
		19	119.7	N 70.5	29.1	66.3	59.5	66.9	66.0
		16	135.7	N 76.1	28.9	67.2	61.8	86.4	84.0
II	Simulated Glass Temperature	19	119.7	N[a]70.5	29.1	66.3	59.5	66.9	66.0
		20	119.8	N 75.2	38.4	66.0	63.0	63.9	62.0
		21	120.2	N 76.1	46.5	65.7	65.0	61.9	61.0
III	Simulated Glass and Ceiling Temperatures	26	129.1	63.0	27.8	N[a]73.8	59.9	76.7	77.5
		27	110.9	63.3	27.2	N 68.3	57.5	56.7	56.0
		28	142.1	65.1	33.0	N 78.6	63.9	91.4	91.0

[a] Neutral surface (neither heated nor cooled).

air. For example, a heat flow rate of 10 Btu per (hr) (sq ft) and a surface temperature of 65 F could represent combinations of wall conductance and outdoor air temperature respectively of 0.5 Btu per (hr) (sq ft) (F deg) and 45 F, 0.20 and 15 F, or any combination in which the product of conductance and temperature difference equaled 10 Btu per (hr) (sq ft).

Effect of wall conductance on the AUST and on the ceiling temperature and heat flow required to maintain an air temperature of 70 F at the 60 in. level is shown in Fig. 10 for conditions of zero F outside air and one air change

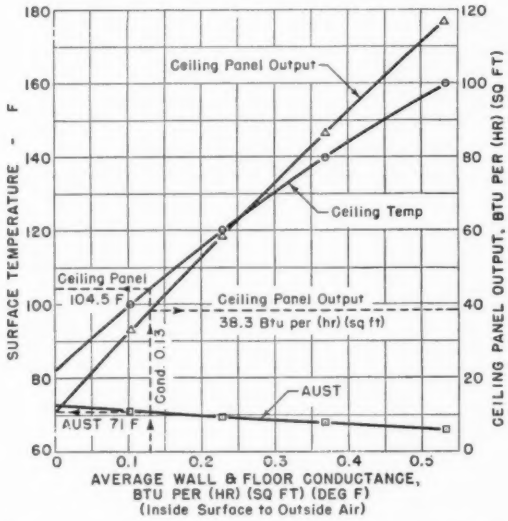


FIG. 10. THE EFFECT OF WALL CONDUCTANCE ON THE AUST AND ON THE CEILING TEMPERATURE AND OUTPUT REQUIRED TO MAINTAIN 70 F ROOM AIR TEMPERATURE AT THE 60 IN. LEVEL

0 outside, one air change at 0 F (2½ x 12 x 8 ft high room)

based upon air density at zero F. The figure illustrates one combination of values. For values of C' greater than about 0.20 the AUST is below 70 F and heat flow is from the air to the walls. For values of C' less than 0.20 the opposite is true. Fig. 10 applies only to the room and the conditions noted.

TEST RESULTS—NON-UNIFORM ENVIRONMENT

Tests were made with various combinations of floor and wall temperatures to explore the performance of a ceiling panel with a non-uniform environment. Results of these tests are shown in Table 4. In Series I the ceiling temperature was the variable while the temperatures of the east wall (simulating glass area) and the other walls were held constant. The floor was a neutral surface, allowed to seek its own temperature level. In Series II the glass temperature was the

controlled variable and the floor a neutral surface. In Series III both ceiling and glass temperatures were the variables. The floor was held at a fixed temperature while three walls were neutral surfaces.

The ninth column of Table 4 gives the ceiling heat flow rates obtained under the various non-uniform test conditions. The last column gives the heat flow rates for the same AUST but for uniform surroundings. These values were obtained from Fig. 1. For these particular tests the agreement is very good and

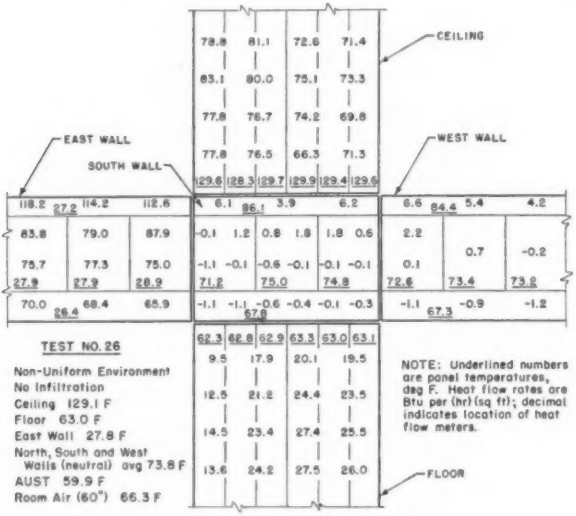

CEILING
EAST WALL
SOUTH WALL
WEST WALL
FLOOR
TEST NO. 26
Non-Uniform Environment
No Infiltration
Ceiling 129.1 F
Floor 63.0 F
East Wall 27.8 F
North, South and West Walls (neutral) avg 73.8 F
AUST 59.9 F
Room Air (60") 66.3 F
NOTE: Underlined numbers are panel temperatures, deg F. Heat flow rates are Btu per (hr)(sq ft); decimal indicates location of heat flow meters.

FIG. 11. HEAT FLOW AND TEMPERATURE DISTRIBUTION FOR SOUTH HALF OF ENVIRONMENT LABORATORY

Non-uniform environment, no infiltration

suggests that AUST may be a satisfactory term to describe a non-uniform surrounding.

The results of Test No. 26 are given in more detail in Table 5 and Fig. 11 and bring out several important points.

(1) The temperatures of the neutral surfaces are well above the temperature of the ambient room air. These temperatures result from an equilibrium condition such that the heat gain by radiation exchange with the ceiling equals the heat lost in the radiation exchanges with the colder floor and east wall plus convection loss to the cooler air. The last column of Table 5 gives the magnitude of the calculated net radiant energy exchange between any one surface and the rest of the room. This is expressed as a percentage of the calculated total radiant output of the ceiling. If reflections are neglected, these fractions hold for any emissivity; the differences between them and the measured heat flow, expressed as a percentage of measured ceiling heat flow, are accounted for by convection exchange with the air.

(2) These percentages show that the east wall received energy both by convection from the air and in the net radiation exchange with the other surfaces. However, both

the floor and the neutral surfaces lost heat to the air by convection and at the same time gained heat in the net radiant exchange with the other surfaces. Moreover, the convection loss from the three neutral walls is just balanced by the radiation exchange, since the net heat flow through these surfaces is practically zero.

(3) It will be noted that in Fig. 11 heat flows into the room from the lower part of the neutral surfaces and out from the room through the upper part. The temperature distribution over the neutral surfaces is roughly what might be expected for inside walls of a normal room.

As previously shown in Table 4 the heat flow rates from a ceiling in a non-uniform environment were not greatly different from those found in a uniform environment with the same AUST. By contrast the air temperatures are found to be different. This is illustrated by Table 6 for three different tests. Fig. 6

TABLE 5—HEAT FLOW DISTRIBUTION FOR NON-UNIFORM ENVIRONMENT WITHOUT INFILTRATION

Test No. 26 (AUST: 59.9 F)

Air Temperature at Center of Room: 58.6 F at 30-in. level; 66.3 F at 60-in. level

SURFACE	SURFACE TEMP F	TOTAL HEAT FLOW BTU/HR	PERCENTAGE OF CEILING HEAT FLOW	HEAT FLOW BTU/(HR) (SQ FT)	ANGLE[a] FACTOR	PERCENTAGE[b] RADIANT ENERGY EXCHANGE
Ceiling	129.1	23,080	100.0	76.7		100.0
Floor	63.0	6,290	27.2	20.9	0.425	38.9
North (incl door)	N[c]74.2	10	0.0	0.1	0.090	3.0
South	N 73.7	75	0.3	0.8	0.090	3.1
West	N 73.4	150	0.7	0.8	0.197	7.0
East	27.8	15,620	67.7	79.7	0.197	48.0
Total:						
Walls and Floor		22,145	95.9			
Unaccounted for		−935	−4.1			

[a] A geometrical factor which takes into account both the shape and the relative position of the surfaces. It represents, as a fraction, the radiant energy emitted from the ceiling which is intercepted by each surface, with no allowance made for reflections. See Reference 4.

[b] This is the calculated net radiant energy exchange between each surface and the rest of the room. Calculation based on emissivity = 1.0.

[c] Neutral surface (neither heated nor cooled).

shows the effect of the non-uniform environment on the air temperature gradient for two different tests. In one case the gradient is quite similar to that for a uniform surrounding, while in the other, the gradient is markedly different.

EFFECT OF ROOM FURNISHINGS

The air temperatures, as found in the tests with a uniform environment, were much lower than normally experienced in practical installations. Several tests were made with sheets of rigid insulation board placed in the room. This provided additional surface area such as would be provided by room furnishings. Four foot high sheets were set vertically, parallel to the four walls, 2½ ft from the walls with about 1½ in. between the floor and the bottom edges of the sheets. The sheets formed a continuous hollow rectangular enclosure with an opening of 18 in. for access.

This arrangement caused an increase in air temperature of about 5 deg for 120 F ceiling and 65 F AUST, both without infiltration and with two air

changes at zero and 40 F. The ceiling output was reduced slightly, heat flow through the floor decreased and all the walls showed an increase in heat flow.

The effect of room furnishings was still further explored in a series of tests with non-uniform environment and with living room furniture. The east and south walls were cooled to simulate the outside walls of the room. Three 4 ft by 5 ft panels in these walls, two in the east wall and one in the south wall, were chilled to a still lower temperature to simulate windows. The other two walls and the floor were neutral and the air temperature back of these areas was held at about 65 F. In these tests, temperatures of the simulated cold walls and glass were held constant while the ceiling temperature was adjusted until the

TABLE 6—EFFECT OF NON-UNIFORM ENVIRONMENT ON ROOM AIR TEMPERATURE WITH NO INFILTRATION

Test No.	Ceiling	Room Surface Temperatures, Fahr				Room Air Temperatures F Deg at 30- and 60-in. Levels			
						Non-Uniform		Uniform	
		AUST	Floor	Glass (East Wall)	Other Walls	30-in.	60-in.	30-in.	60-in.
26	129.1	59.9	63.0	27.8	N[a]73.8	58.6	66.3	62.6	65.4
27	110.9	57.5	63.3	27.2	N 68.3	57.8	62.9	59.6	61.8
28	142.1	63.9	65.1	33.0	N 78.6	62.2	71.1	66.9	70.1

[a] Neutral surface (neither heated nor cooled).

room air temperature at 60 in. level was substantially 70 F. Tests were made as follows:

A. Non-uniform environment, empty room,

B. Non-uniform environment with furniture,

C. Non-uniform environment with furniture and lights (900 watts),

D. Non-uniform environment, with furniture, lights (900 watts) and drapes covering the simulated windows.

In each case tests were made without infiltration and with one air change at 20 F outside temperature. The furniture consisted of a settee, five upholstered chairs, one large table, two end tables, two torch-type floor lamps, and one table lamp.

Fig. 12 shows the temperature and heat flow rate distribution for the empty room without infiltration. The effect of the cold windows on the heat flow through the wall sections below them will be evident.

The relationship of these test conditions to practical situations can be shown by assuming that the outside air temperature is 20 F for the conditions shown in Fig. 12. For the average of the heat flows shown for the *glass,* the window conductance (inside glass surface to outside air temperature) is 4.3 Btu per (hr) (sq ft) (F deg). This corresponds to an overall coefficient, air-to-air, *U* of 1.19 Btu per (hr) (sq ft) (F deg) when the commonly used inside coefficient of 1.65 Btu per (hr) (sq ft) (F deg) is applied. On the same basis the south wall has a *U* value of 0.14 and the east wall a *U* value of 0.16.

Ceiling temperatures required to maintain approximately 70 F room air temperature and the corresponding heat flow rates are given in Table 7. Also listed

are the AUST values. As furniture, lights, and drapes hung over the windows were progressively added, the ceiling temperature required, and the accompanying heat flow rate, were reduced. A change in AUST reflects the change in temperature of the neutral floor and the two neutral walls as the test conditions were changed.

Comparison of Panel Performance in Uniform and Non-Uniform Environment: In Table 7 panel output and room air temperature for each of the eight tests are compared with the estimated values for an empty room with a uniform environment which has the same AUST and ceiling temperature. These values

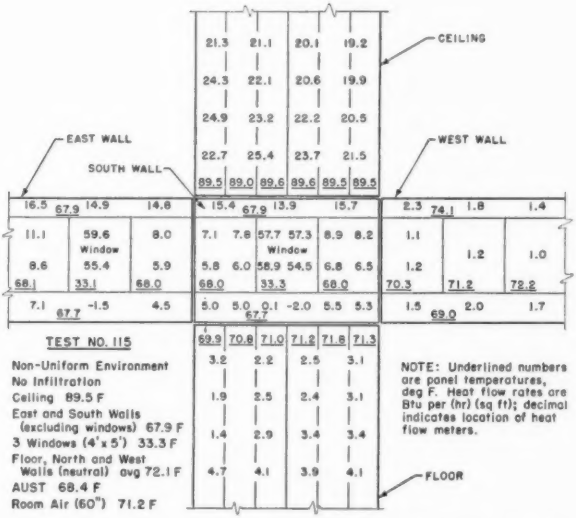

CEILING
EAST WALL
SOUTH WALL
WEST WALL
FLOOR
21.3 21.1 20.1 19.2
24.3 22.1 20.6 19.9
24.9 23.2 22.2 20.5
22.7 25.4 23.7 21.5
89.5 89.0 89.6 89.6 89.5 89.5
16.5 67.9 14.9 14.8
11.1 59.6 Window 8.0
8.6 55.4 5.9
68.1 33.1 68.0
7.1 67.7 -1.5 4.5
15.4 67.9 13.9 15.7
7.1 7.8 57.7 57.3 Window 8.9 8.2
5.8 6.0 58.9 54.5 6.8 6.5
68.0 33.3 68.0
5.0 5.0 0.1 -2.0 5.5 5.3
67.7
2.3 74.1 1.8 1.4
1.1 1.2 1.0
1.2
70.3 71.2 72.2
1.5 69.0 2.0 1.7
62.9 70.8 71.0 71.2 71.8 71.3
3.2 2.2 2.5 3.1
1.9 2.5 2.4 3.1
1.4 2.9 3.4 3.4
4.7 4.1 3.9 4.1
TEST NO. 115
Non-Uniform Environment
No Infiltration
Ceiling 89.5 F
East and South Walls (excluding windows) 67.9 F
3 Windows (4'x 5') 33.3 F
Floor, North and West Walls (neutral) avg 72.1 F
AUST 68.4 F
Room Air (60") 71.2 F
NOTE: Underlined numbers are panel temperatures, deg F. Heat flow rates are Btu per (hr) (sq ft); decimal indicates location of heat flow meters.

FIG. 12. HEAT FLOW AND TEMPERATURE DISTRIBUTION FOR SOUTH HALF OF ENVIRONMENT LABORATORY

Simulated room conditions, no furnishings

were taken from Figs. 1 and 4. Heat flow in the A tests is very nearly the same as the heat flow for a uniform environment. This was also observed in other tests (see Table 4). The results of the B tests in which furniture was added show little change in this relationship even though the furniture shields some surfaces from the ceiling panel. In the C tests the heat outputs were substantially lower because of the addition of lights, the heat input from which amounted to about 10 Btu per (hr) (sq ft of ceiling area). The difference in panel outputs, compared to the corresponding uniform environment values, is only about one half of the energy input from the lamps. The addition of drapes over the windows caused a somewhat greater difference between heat flows in the D tests and the corresponding uniform environment values.

In each case the estimated room air temperature is lower for the empty room with uniform environment than the observed value in the tests with

non-uniform environment and furniture. The values shown were taken from Figs. 7 and 9.

SUMMARY

Air temperatures and heat flow rates in this paper pertain only to the room in which the investigation was made. The greatest part of the investigation has dealt with a uniform temperature environment with the entire ceiling heated. The effects of many important variables remain to be considered. These are

TABLE 7—HEAT FLOW RATES AND AIR TEMPERATURES FOR A ROOM WITH NON-UNIFORM ENVIRONMENT COMPARED WITH VALUES FOR AN EMPTY ROOM WITH UNIFORM ENVIRONMENT

TEST CONDITION	TEST NO.	CEILING TEMP F	AUST F	CEILING HEAT FLOW BTU/(HR) (SQ FT)	ROOM AIR TEMP AT 60 IN. LEVEL F
With No Infiltration					
A	115	89.5	68.4	21.2	71.2
Uniform[a]		89.5	68.4	22.0	70.0
B	124	88.3	66.8	20.9	70.4
Uniform[a]		88.3	66.8	22.0	68.6
C	118	83.5	67.0	11.5	70.0
Uniform[a]		83.5	67.0	16.5	68.3
D	120	82.7	67.3	8.3	70.0
Uniform[a]		82.7	67.3	15.5	68.5
With One Air Change at 20 F					
A	116	106.7	70.7	39.9	70.9
Uniform[a]		106.7	70.7	40.8	70.4
B	123	101.7	67.9	35.5	70.2
Uniform[a]		101.7	67.9	36.6	67.6
C	117	97.1	68.9	24.9	70.2
Uniform[a]		97.1	68.9	30.3	68.1
D	119	93.9	68.3	20.5	70.2
Uniform[a]		93.9	68.3	27.1	67.2

A—Non-uniform environment, empty room.
B—Non-uniform environment, furniture.
C—Non-uniform environment, furniture and lights (900 watts).
D—Non-uniform environment, furniture, lights (900 watts) and drapes.

[a] Uniform environment, empty room (estimated).

occupancy, points of entrance and exit of infiltration air and non-uniformity of temperature of actual exposed walls.

The effect of room size on panel performance has not been investigated. The construction of the Environment Laboratory is such that ceiling height can readily be changed and that two smaller rooms can be made from the present large one. The many possible room dimensions that the designer may encounter suggest that a simple approach may be through evaluation of the convective and radiant modes of heat transfer.

However, *for this room and the specific tests conditions reported here* the following observations can be made:

1. The total heat output of the ceiling panel was much lower than is given in some design methods now used. This seems to be due principally to the fact that the con-

vection conductances obtained for the ceiling panel appear to be much lower than presently published data would indicate.

2. The heat flow due to radiant exchange between a given surface and the rest of the room may be opposite in direction to the heat flow due to the convective exchange between the surface and the ambient air.

3. The surface temperatures of neutral walls were not necessarily the same as the temperature of the ambient air but were dependent upon the heat balance between the radiative and convective heat exchanges.

4. For this room, a definite relationship was established between room surface temperatures, infiltration air rate and temperature, and the temperature of the room air at the 60 in. level.

5. With no infiltration the AST minus the room air temperature was directly proportional to the ceiling temperature minus the AUST. For constant values of ceiling temperature and AUST, the difference between the AST and room air temperature increased as the rate of infiltration increased and as the temperature of the infiltration air decreased. As defined before, the AST is the area-weighted average temperature of *all* the room surfaces, and the AUST is the area-weighted average temperature of the *unheated* surfaces of the room.

REFERENCES

1. The A.S.H.V.E. Environment Laboratory, by Cyril Tasker, C. M. Humphreys, G. V. Parmelee and L. F. Schutrum (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March 1952, p. 119).

2. A.S.H.V.E. RESEARCH REPORT No. 1453—A Low-Inertia Low-Resistance Heat Flow Meter, by R. G. Huebscher, L. F. Schutrum and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 275).

3. A.S.H.V.E. RESEARCH REPORT No. 1402—A Simple Heated-Thermocouple Anemometer, by H. B. Nottage (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 431).

4. *Panel Heating and Cooling Analysis*, by B. F. Raber and F. W. Hutchinson (John Wiley and Sons, Inc., New York, 1947).

DISCUSSION

W. P. CHAPMAN, Pittsburgh, Pa., (WRITTEN): What was the magnitude of gaseous absorption of radiation? It was mentioned that, *Dew point temperature had a measurable affect on room air temperature because of absorption.*

Prof. F. W. Hutchinson, in a paper, theorized that gaseous absorption is 10 percent in an *average* room under *average* panel heating conditions.* With metallic wall surfaces the 10 percent would be increased. Here is an excellent opportunity to test the equations developed in his paper. This would be valuable information.

In the third method used to calculate the infiltration load, the exfiltrating air approached the AST. In the past, it has been assumed that exfiltration occurred at room air temperature. Since in usual conditions the AST is lower than the air temperature, it seems that infiltration loads can easily be reduced. This point certainly deserves further consideration, especially since exfiltration generally occurs around doors and windows in a far greater proportion than in any other way.

Two values were given for the convection coefficient: 0.80 and 0.04. This is a 20 to 1 ratio. As the authors imply, the probable coefficient for practical application will be somewhere between these two extremes. The only way to clarify this problem

* Influence of Gaseous Radiation in Panel Heating, by F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 285).

is to carry out tests for a range of air velocities and then compare the test velocities to actual field velocities.

This one paper has opened, or reopened, four fields of investigation: (1) the magnitude of gaseous absorption in a panel heating system, (2) the determination of the temperature of exfiltrating air, (3) the determination of the variation of convection with air velocity, and (4) the determination of the most probable air velocity to be found in a panel heated room.

J. R. CARROLL, Urbana, Ill., (WRITTEN): This paper presents data on the total heat transfer to a uniform environment in which a ceiling panel is contained within the confines of the walls. It also shows there is less heat transfer than theoretical rates based upon the extrapolation of data obtained from research on small and relatively unrestricted high temperature sources such as wires, spheres, cylinders, and vertical plates. These are the only data previously available. The authors mention that an apparent difference does exist, and they present $h_1 = 0.04$ Btu per (hr) (sq ft) (F deg) as representing the unit convection conductance. This free convection coefficient is based upon the observed air temperature difference between the air at the 60 in. level in the room center and the average ceiling surface. It would have been a different number if the air temperature were taken at some other level, due to the rapid increase of air temperature gradient close to the ceiling because of the confinement of the ceiling within the room.

When operated with a fan blowing at the ceiling, the coefficient h is increased to 0.80 and the air temperature at the 60 in. level at the center of the room also increased to 80.7 F. Since the radiation exchange was essentially the same for the two tests from which these two coefficients were derived, the calculated radiation component is approximately 56.2 Btu per (hr) (sq ft). By applying this same radiation component to the data of Table 2, which include infiltration and which also have essentially the same surface temperatures and, therefore, the same radiation exchange as the other data, it is found that $h = 0.14$. This increase compared to $h = 0.04$ is due to the lower room air temperature at the 60 in. level and the higher rate of heat transfer because of the infiltration air and serves to illustrate that h is not a constant but varies with application, and since an actual application includes infiltration, it is desirable to know this variation.

This paper does not develop the relationship between free convection heat transfer and temperature difference directly, but if the data are available, it should be accomplished in order to fill in a large gap in heat transfer history. All of the previously reported work has been done with vertical surfaces and small objects in which the air flow is unimpeded.

Since it appears as if the data may now be available from this excellent work at the A.S.H.V.E. Laboratory, this basic heat transfer relationship should be determined for the ceiling position.

D. L. MILLS, Rome, N. Y. (WRITTEN): The data given in the paper pertain only to the particular room in which the investigation was made. This is very important because of the tendency to use data obtained in a laboratory for designs affected by field conditions that may be very different from those in the laboratory.

Even if no other technical papers were published, this paper, alone, shows the great potential value of the Environment Laboratory. Test room surfaces can be held to close desired values, and accurate energy balances.

The summary states that the convection conductances obtained from the ceiling panel appear to be much lower than currently published data would indicate. Can a safe convective film coefficient be devised for design purposes in the average residence?

Although uniform surface temperatures and an absence of infiltration affect heat transfer by convection from a ceiling panel, the actual rate of this transfer, under such

conditions, can be determined. Of interest is the influence of air movement over a heated panel on the heat output from it.

H. B. NOTTAGE, Los Angeles, Calif., (WRITTEN): Thermal circuits for air conditioning have been under study recently at the A.S.H.V.E. Research Laboratory, and a paper is in preparation. It is suggested that the authors present and condense their results by this method. For example, Fig. A shows electrical circuits representing

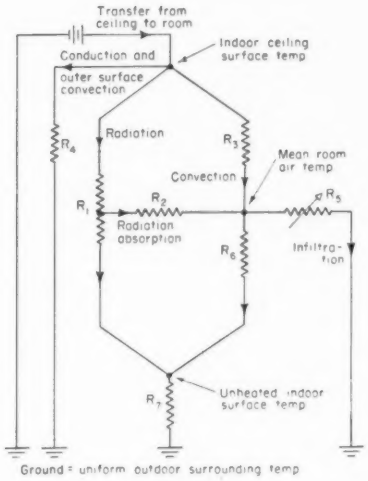

Transfer from ceiling to room
Indoor ceiling surface temp
Conduction and outer surface convection
Radiation
Mean room air temp
Convection
Radiation absorption
Infiltration
Unheated indoor surface temp
Ground = uniform outdoor surrounding temp
Resistances
R1 = radiation interchange
R2 = equivalent of radiation absorption
R3 = ceiling to air convection
R4 = inner ceiling surface to outdoor surroundings
R5 = equivalent of infiltration
R6 = air to unheated surface convection
R7 = unheated surface to outdoor surroundings

FIG. A. ELECTRICAL CIRCUIT REPRESENTATION OF THERMAL CONDITIONS IN ENVIRONMENT LABORATORY (FOR UNIFORM HEATED CEILING AND UNIFORM UNHEATED SURFACES)

thermal conditions in the Environment Laboratory under the condition of uniform ceiling and unheated surface temperatures.

The potentials, resistances, and currents in this system are interrelated by standard circuit equations. Some complication of detail can arise from the fact that resistances and potentials may be interdependent, although simplifying approximations can easily be justified in practical usage.

The more complete case of different temperatures of the various room surfaces is represented by a circuit having additional branches.

Circuit techniques, supported by valid data on resistances, will enable the authors

to treat more properly the following items in their paper: (1) Gaseous radiation, (2) Limitations on average surface temperature, AST, (3) Limitations on average unheated surface temperature, AUST, (4) Relation between air temperature, ceiling temperature, and average unheated surface temperature, AUST. (5) Reflected radiation, (6) Relation between AUST and a more complete description of non-uniform surroundings, (7) Furnishings and sources within the room.

F. E. GIESECKE, New Braunfels, Tex. (WRITTEN): The chart of Fig. 1 is very similar to the theoretical chart published in 1940.* For a ceiling temperature of 100 F, the chart of Fig. 1 shows a *total heat flow* of 50, 41, 37, and 33 with an average of 40.25 Btuh per sq ft when the average temperatures of the five unheated surfaces are, respectively, 50, 60, 65, and 70 F. For the same temperature differences, the 1940 theoretical chart shows a *total radiant heat flow* of 48, 39, 35, and 31 with an average of 38.25 Btuh per sq ft.

If the two charts are comparable, the *total convected heat flow* is 40.25 less 38.25 Btuh per sq ft or about 5 percent of the total radiant heat flow, then the ceiling is the heating panel.

Some such simple relation of *convected* heat flow to *radiant* heat flow, for the several possible locations of the heating panel, would be of value to the designer.

The 1940 theoretical chart assumed that the ceiling heating panel was completely enclosed by the floor and the four walls so that the radiant heat flow is 0.156 $[(T_1/100)^4 - (T_2/100)^4]$ Btuh per sq ft if the mean emissivity of the surfaces is 0.9 and if T_1 is the average absolute temperature of the heated surface and T_2 is the area weighted average absolute temperature of the unheated surfaces which enclose the heated surface.

It should be determined whether this approximate method of calculating the radiant heat flow is sufficiently accurate for design purposes, for rooms of ordinary size and ordinary proportions.

R. J. MINDAK, Chicago, Ill.: The total ceiling heat output of approximately 92.5 percent by radiation and 7.5 percent by convection, established in this paper, is quite different from the old theory that 67 percent of the output is due to radiation. These results indicate more data are needed, covering free convection heat flow from large horizontal surfaces.

Members of the profession are interested in the design of radiant ceilings as well as the accuracy of design calculations in this paper. There are some data presented comparing rooms of uniform and non-uniform environment, in which little difference in output was noted. It is doubtful if the small difference in output between rooms of uniform and non-uniform environment will exist for every case. For example, what is the radiant heat flow between the ceiling and the window in a room 12 x 20 x 9 ft high with a large picture window 20 x 9 ft high? Assume all other surfaces as adiabatic, an overall AUST of 60 F, window surface temperature of 33 F, with 76 F room air temperature and an average ceiling surface temperature of 104 F. Using the A.S.H.V.E. methods, which neglect the effect of geometrical configuration, the calculated output is approximately 43 Btuh per sq ft. Using the methods of Hottel, accounting for geometrical configuration and assuming adiabatic surfaces are reradiating and reflecting, the calculated output is approximately 73 Btuh per sq ft. Therefore, for the case of a room of non-uniform environment having a very large and very cold wall, radiation calculations which neglect geometrical configuration may give results which are quite different from the actual case. More data are needed to either confirm or deny this. Fortunately, however, any existing errors are on the conservative side.

---

* Radiant Heating and Cooling, by F. E. Giesecke (*Heating, Piping & Air Conditioning*, August 1940, p. 484).

The data inserted in this paper approach those obtained in similar tests at Armour Research Foundation.

A. T. JORN, Chicago, Ill.: Tests at Armour Research Foundation produced results very comparable to the data reported in this paper. The ARF test room was operated with just one cold wall, but the results were very similar to those from the authors' test room which had a uniform surface temperature.

Heating engineers realize that, in practice, both conventional embedded-coil and the newer non-embedded-coil panel ceilings have non-uniform surface temperatures, consequently the surface directly below the coil tubes is warmer than the surface mid-way between coil tubes. How the uniform surface temperature data previously provided should be applied to the practical design situation has long been a question.

ARF tests indicate that a simple arithmetic average measurement of panel surface temperature, taken at close intervals from a point directly below one coil tube and on across the intervening surface to a point below the next coil tube, will obtain an *effective panel temperature*. This temperature may be used on the uniform surface temperature charts to secure the correct panel heating capacity rating. This procedure, however, is not entirely accurate. If the tubes are at a great enough spacing, or if the intervening material is such a poor conductor that the panel surface mid-way between coil tubes is at a temperature approaching the average unheated surface room temperature, the ARF arithmetic average varies greatly with the actual *effective* panel temperature. Therefore, if the average panel temperature includes the temperature of some area that is essentially at room temperature, the actual output will vary greatly from the value indicated by the average surface temperature.

Recent panel cooling tests indicate that ceiling panel cooling capacities can be approximated closely by applying conventional formulas as provided in THE GUIDE 1952. That the data should evolve this way is rational, for in both floor panel heating and ceiling panel cooling the air at panel temperature is moved away from the panel by the same gravitational or natural convection force. The radiation exchange is, of course, independent of any inversion of the geometrical relationship.

This paper indicates that the Nusselt and Henky equation used to evaluate heat transfer by natural convection from a ceiling heating panel, is in error. This is probably due to not confining convection from the edges of the free-hanging panel in the Nusselt and Henky experiments, as it would be confined by walls in normal field installations.

P. B. GORDON, New York, N. Y.: Some extensions to the data in this paper have been made for rooms of other sizes and other shapes. However, this is not measured, but calculated data. The deviation is not too serious. The Environment Laboratory provides this facility, provides the tool to permit checks on these variations.

It is expected that the next six or eight months will produce additions to the work that has just been completed, information that will provide answers to questions raised in this paper. This includes information on floor panel outputs, similar to the work on ceiling panel outputs for rooms of uniform environments. Further, a whole series of studies are being carried forward on the problem of non-uniformity, including the differences caused by walls and large picture windows, and cases wherein sections of the ceiling or floor are heated.

Additional extensive work is also being carried forward to determine surface conductances for the different heating and receiving surfaces, including accurately measured emissivities of the different surfaces. This includes study of the effects of room furnishings.

P. R. ACHENBACH, Washington, D. C.: National Bureau of Standards tests, made shortly before the experiments noted in this technical paper, compare favorably. Total heat output comparisons for any given ceiling temperature and AUST for the different pieces of test apparatus are within about 2 Btu per sq ft of surface. Although

these data could be applied to actual houses as far as total heat transfer is concerned, conclusions reached about the proportions of heat transferred by radiation and convection would not be entirely accurate.

In uniform environment without infiltration, about 42 percent of the heat output from the ceiling was transmitted by the floor. In a more or less typical house construction, the heat transfer through the floor would be on the order of 10 or 15 percent of the total heat loss of the building. In the Environment Laboratory all the radiant heat received was transmitted, or practically all of it, because the air was either warmer than or at the same temperature as the room surfaces. In a house where the floor was transmitting only some 15 percent of the heat, it would be warmed above the air temperature and convection might occur at the floor and considerably increase convection heat transfer from the ceiling in a non-uniform environment.

The tests made with a non-uniform environment show a lesser amount of heat transferred by the floor but it still exceeds the percentage of the total heat loss transferred by the floor in most houses.

AUTHORS' CLOSURE (C. M. Humphreys): Gaseous absorption of radiation changed during Environment Laboratory tests. The effect of dew-point temperature was listed in the technical paper to establish this fact. Tests were made in winter at rather low dew-point conditions. Results of several checks at higher dew-points did not seem to correlate well. Special experiments proved that the differences in observed air temperatures were due to greater absorption of radiant energy by water vapor at higher dew-point conditions. Accurate evaluation of this factor was not attempted, but in one case, an increase in dew-point from zero to 60 F, with AUST and panel temperature held constant, caused an increase in room air temperature of approximately 3.5 F.

Metallic wall surfaces would increase the amount of radiant heat absorbed by water vapor if the surfaces were reflective. The Environment Laboratory metal walls are covered with several coats of a typical interior paint to diminish absorption phenomena in the metal to an extent equal to the average structure.

Changing calculation of infiltration losses would be unwarranted application of data developed in the Environment Laboratory. In a normal 12 x 24 ft room, there is a comparatively small amount of crack for either infiltration or exfiltration.

Environment Laboratory infiltration air conditions were quite different. More than 1500 linear ft of crack are between the metal panels which form room surfaces. Air entered the room through perforated pipes, and approximately 50 percent of it left the room by slowly seeping through the cracks in the six-room surfaces. A heat balance indicated an exiting air temperature approximately the same as the room's average surface temperature. This excessive amount and unusual distribution make it doubtful if the same relation between temperature of room surface and outgoing air would hold for a normal heated structure.

Air might exfiltrate from a room at a different temperature than that of its passageway. In a normal building the window crack temperature is certainly well below the AST; therefore, the air should not be expected to leave the room at the AST. A multi-room structure is quite likely to have air infiltrating into a given room leaving it through an open door to corridor or other room, rather than through door or window cracks.

Additional data are needed on convection heat transfer. Work is now in progress to determine the radiation and convection components of the heat exchanges. Results will be reported in a later paper.

The thermal circuit method of analysis should prove helpful in solving many heating and air conditioning problems. Work on the application of this method of analysis is under way at the Laboratory.

Examples have been cited of close agreement between other experiments and Environment Laboratory tests. Further tests are needed with non-uniform environments.

1474

No. 1474

THE SHADING OF SUNLIT GLASS

An Experimental Study of Slat-Type Sun Shades

By G. V. PARMELEE*, W. W. AUBELE**, AND D. J. VILD†, CLEVELAND, OHIO

THIS PAPER gives the results of experimental studies of the effectiveness of slat-type shades on reducing heat gain through sunlit glass. The investigation was carried on at the A.S.H.V.E. Research Laboratory as part of the long-range research program of the Technical Advisory Committee‡ on Heat Flow Through Glass.

A previous paper[1] on this subject gave a mathematical analysis by which the absorbed, reflected, and transmitted fractions of solar radiation incident upon a shade-glass combination could be computed. The present paper compares experimental determinations of the absorbed and transmitted fractions with the mathematical theory. Experimentally determined relationships between the heat gain by convection and re-radiation and the amount of solar radiation absorbed by the shade-glass combination will be covered in a later paper. Correction factors for shading effects to be applied to heat gain data for unshaded glass can be determined from these basic results; they will be the subject of still another paper.

SCOPE OF EXPERIMENTS

In the slat-type sun shades of which venetian blinds, sun screens, and jalousies are common examples, there are many possible combinations of variables. These

* Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

** Research Engineer, A.S.H.V.E. Research Laboratory. Now with Florida Power & Light Co., West Palm Beach, Fla.

† Research Engineer, A.S.H.V.E. Research Laboratory.

‡Personnel: R. A. Miller, *Chairman;* N. B. Hutcheon, *Vice Chairman;* W. J. Arner, F. L. Bishop, E. W. Conover, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, D. R. Muir, H. B. Vincent, J. R. Young.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

experiments were restricted to four shades, but these four gave reasonable coverage to the most common applications. The range of variables studied is shown in Table 1.

TEST APPARATUS

Heat gain measurements were made by means of the solar calorimeter previously described.[2, 3] Where the slats were installed on the outdoor side of the glass, a light black-painted wood frame so constructed as to avoid partial shading of the glass by the frame held the slats at the desired angle and spacing. The closest edge of the slats was ¾ in. from the glass.

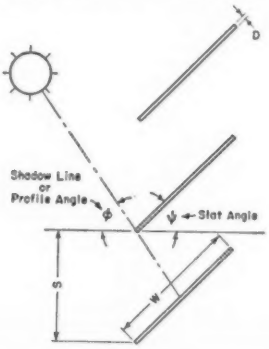
Slats installed inside the calorimeter were supported by black painted wood strips fastened to the metal surface of the calorimeter (see Fig. 2). The front

TABLE 1—RANGE OF VARIABLES STUDIED

1. Shade in Combination with Glass No. 2 or No. 9[a] (Transmittance of 90 percent for normally incident direct solar radiation[2]) under following conditions:

Slat Width-Spacing Ratio, W/S:	0.6, 0.8, 1.2
Slat Width, W:	0.047, 2.00 in.
Slat Thickness-Spacing Ratio, D/S:	zero (nominal), 0.025, 0.085
Slat Angles, ψ:	20, 30 and 45 deg
Slat Orientation:	Long dimension of slats horizontal
Diffuse Reflecting Slats:	Total solar energy absorptances, α, of approx. 0.3, 0.6, 0.8 for white, green, and dark colored respectively.
Specular Reflecting Slats:	Total solar energy absorptance, α, of 0.16 (aluminum)
Shadow Line or Profile angle, ϕ:	0-75 deg (see Fig. 1)
Indoor Environment:	Black body at approximately 75 F
Outdoor Environment:	Clear days in spring, summer and fall
Shade Location:	Shades both indoors and outdoors; plane of shade assembly parallel to plane of glass

2. Shade in Combination with Glass No. 6[a] (Transmittance of 41 percent for normally incident solar radiation[2]) with white shade indoors with $W/S = 1.2$, $\psi = 30$ deg.



[a] Glasses described in Reference 2.

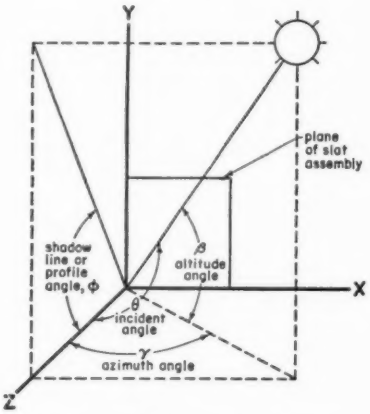


FIG. 1. DEFINITION OF ANGLES

edge of the slats was 13/16 in. back of the glass. This set-back and the support strips caused the true shaded area to be somewhat less than the clear opening of the glass. Tapes, cords and slat tilting mechanisms were omitted in all tests.

The white and green slats were standard venetian blind slats of slightly curved 2-in. wide steel strips with baked enamel paint. The dark slats were a venetian blind type of sun and insect screen of chemically treated bronze. The aluminum slats were made by cementing together two 2-in. wide flat strips of specular reflecting lighting sheet.

Temperatures of the green and white slats were measured by thermocouples flattened and cemented to the slat surfaces. The couples for the aluminum slats were inserted between the two strips making up the slat. Those for the screen-type shade were soldered to the slats.

TEST PROCEDURE

Six different conditions for tests were used to obtain the desired information on solar energy absorptance and transmittance of slat-type shades as follows:

A. Shade Installed Outside the Glass

1. Calorimeter in a vertical position facing away from the sun. Shade and glass exposed only to diffuse solar radiation.

2. Calorimeter exposed to the direct sun; required profile angle maintained by continuous adjustment of the tilt and azimuth of the calorimeter; most tests made with zero azimuth with respect to the sun.

3. Calorimeter set in a vertical position with a fixed orientation.

B. Shade Installed Inside the Glass

1. Same as in A-1.

2. Calorimeter placed in a vertical position and exposed to direct sun; continuously rotated to maintain zero azimuth with respect to the sun.

3. Same as in A-3.

Tests for conditions A-1, A-2, and B-1 usually approximated steady-state conditions in that there were relatively small changes in solar intensity and air temperature for the duration of the test periods, which usually were ½ hr in length. Tests under conditions A-3, B-2, and B-3 were made primarily to obtain data by which convection and radiation gain could be correlated with the amount of solar radiation absorbed, and with temperature difference, wind velocity, and location and type of shade. Table A-1 (Appendix A) gives typical test data

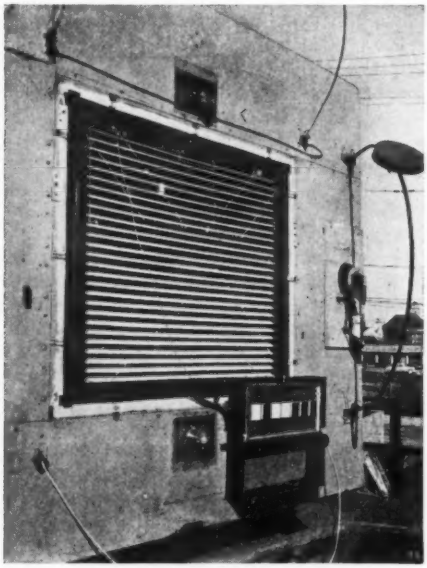


FIG. 2. INSTALLATION OF WHITE SLAT-TYPE SUNSHADE IN SOLAR CALORIMETER. SAMPLES MOUNTED FOR SOLAR ABSORPTIVITY TESTS ARE ALSO SHOWN

and the sequence of calculations by which transmittance and absorptance values were obtained.

RADIATION CHARACTERISTICS OF THE SLAT SURFACES

Solar Energy Absorptance: The spectral absorptance values for the slats of the four different sun shades studied are shown in Fig. 3. Values for a fifth slat, E, which is used in a venetian blind type of sun and insect screen, are also given. Although values for three of the slats cover only part of the solar spectrum, about 80 percent of the total energy of the direct rays of the sun lies in this part of the spectrum, and calculations based on these values should yield

a fair estimate of the absorptance for the entire solar spectrum. Calculated total solar energy absorptance values are given in the right column of Table 2.

Table 2 also gives total energy absorptance values determined by exposing the slats to direct sunlight. Samples were mounted on a block of cork covered with aluminum foil (see Fig. 2). Thermocouples were attached to the under side of each specimen and to the foil covered cork. The samples were exposed to perpendicularly incident solar radiation, the intensity of which was measured by the pyrheliometer. The incident low temperature radiation was simultaneously measured with a convection compensated radiometer.

Emissivity values of 0.92, 0.70 and 0.06 were assigned to the painted, the specular aluminum, and the aluminum foil specimens respectively. The convection conductance was computed from the observed temperature of a flat-black painted specimen, to which was assigned a value of 0.95 for both emissivity and absorptivity. Absorptance values were then determined by solving the usual heat balance equation.

Because of the foregoing assumptions, the test results can only be regarded as *order of magnitude* values. However, the value for specimen C is in reasonable agreement with the computed value, and the value for specimen B is close to the calculated value, which, as noted, applies to only part of the solar spectrum. The large difference between the test and computed values for specimen A cannot be accounted for. However, data[5] for another white paint which has nearly the same absorptance in the region of 0.45 to 1.15 microns suggest that the absorptance for the entire solar spectrum should be about 0.30. The value of 0.19 for aluminum foil is reasonably close to the value of 0.15 quoted by Edwards[6] for visible light absorptance of commercial foil.

Reflection Characteristics: Specimen C appeared to be a perfect specular reflector. Specimens A and B both had definite specular reflection components and therefore gonio-photometer tests were made on each for light incident at an angle of 7½ deg. The results are shown in Fig. 4. The measurements were made only in the plane which contained both the incident beam and the reflected specular beam.

TABLE 2—TOTAL SOLAR ENERGY ABSORPTANCE VALUES

SPECIMEN	DESCRIPTION	ABSORPTANCE VALUES, α	
		By Test in Sunlight	Calculated from Spectral Data[d]
A	White painted slat[a]	0.37	0.25[e]
B	Green painted slat[b]	0.58	0.55[e]
C	Specular aluminum	0.20	0.16[f]
D	Chemically treated bronze	—	0.80[e]
E	Chemically treated aluminum alloy	—	0.36[f]
F	Same as specimen A except 3 coats paint[c]	0.28	—
G	Aluminum foil	0.19	—

[a] 0.0008 in. baked enamel paint film on a zinc phosphate coating applied to electrolytically galvanized cold rolled steel; titanium dioxide pigment; alkyd urea melamine vehicle.
[b] Same as A except paint tinted with chrome green and chrome yellow.
[c] Same as A with the addition of 3 brush painted coats of titanium dioxide pigment white paint; total thickness of paint film 0.003 in.
[d] Moon's[4] solar intensity values for air mass = 2.
[e] For portion of solar spectrum between 0.45 and 1.115 microns.
[f] For portion of solar spectrum between 0.35 and 2.2 microns.

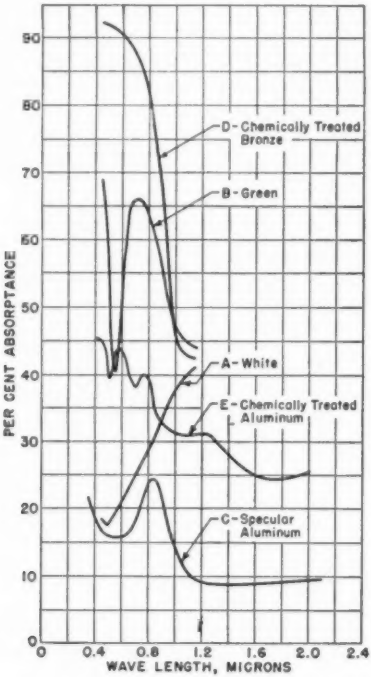


FIG. 3. SPECTRAL ABSORPTANCE OF SUN SHADE SLATS

TABLE 3—DIFFUSE SOLAR RADIATION TRANSMITTANCE VALUES FOR SLAT-TYPE SUN SHADES

SHADE	COLOR	SLAT WIDTH SPACING RATIO, W/S	SLAT ANGLE ψ	TRANSMITTANCE[a]			
				Experimental		Calculated[b]	
				$T_{d,s,g}$		$T_{d,s,g}$	$T_{d,s}$
				Avg	Range		
A	White	1.2	30	0.28	0.27-0.33	0.16	0.19
A	White	1.2	45	0.26	0.23-0.27	0.09	0.10
B	Green	1.2	30	0.17	0.12-0.27	—	0.13
B	Green	1.2	45	0.16	0.13-0.22	0.05	0.06
C	Specular aluminum	1.2	30	0.49	0.39-0.61	0.46	0.52
D	Treated bronze	0.8	20	0.35	0.27-0.41	0.28	—

[a] $T_{d,s,g}$ is the transmittance, for diffuse radiation, of the combination of the shade and glass, with the shade on the outdoor side; $T_{d,s}$ is the transmittance, for diffuse radiation, of the shade alone.

[b] For sky of uniform intensity; no ground reflected radiation; based on slat absorptance values of 0.40 0.60, 0.16, and 0.80 for A, B, C and D respectively. $T_{d,s}$ for shade A with absorptance = 0.30 would be 0.22 and 0.13 for ψ = 30 deg and ψ = 45 deg respectively

If a surface is a perfectly diffusing reflector, the intensity of the reflected beam follows the cosine law with the beam maximum intensity being perpendicular to the surface. The intensity values, plotted *vs.* angle on polar coordinates, will form a circle with the value of maximum intensity as diameter. On this basis the component in the region of the angle of specular reflection was estimated to be of the order of 10 percent of the total reflected light. Measurements of the specular component by allowing the reflected light to leave the

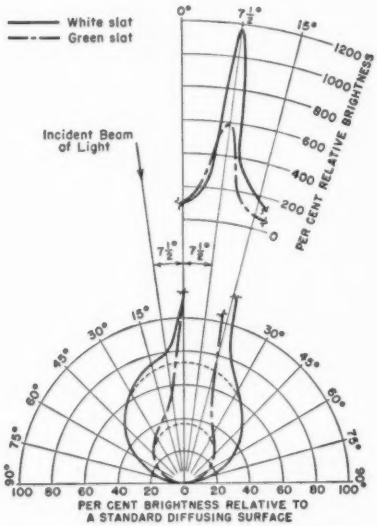

White slat
Green slat
Incident Beam of Light
PER CENT RELATIVE BRIGHTNESS
PER CENT BRIGHTNESS RELATIVE TO A STANDARD DIFFUSING SURFACE

FIG. 4. ANGULAR DISTRIBUTION OF LIGHT REFLECTED FROM WHITE AND GREEN SLATS

integrating sphere of the spectro-photometer gave a value of about 3 percent. Therefore, it appears that the white and green slats reasonably approximate the characteristics of a perfectly diffusing reflector. In the absence of specific data it will be assumed that specimen D slats reflect similarly.

SOLAR ENERGY TRANSMITTANCE AND ABSORPTANCE

Transmittance of Diffuse Solar Radiation: Diffuse transmittance values were determined from data obtained in the condition A-1 tests; the sequence of computations is listed in column 1 of Table A-1. Values for shade-glass combinations, with the shade on the outdoor side are given in Table 3, and are averages of from 4 to 10 tests. Table 3 also gives two sets of calculated values for a sky of uniform intensity (see Reference 1 for calculation method).

The range in the experimental values given in Table 3 is considerable and is partly due to experimental error, since small errors in the various measurements

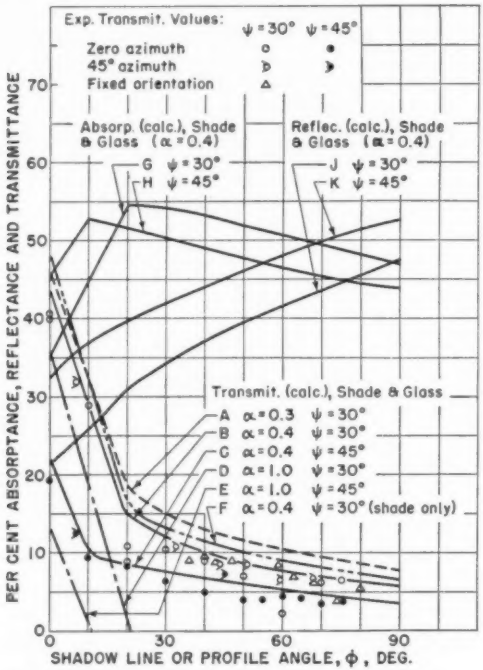


FIG. 5. ABSORPTANCE, REFLECTANCE, AND TRANSMITTANCE OF WHITE SLAT-TYPE SUN SHADES IN COMBINATION WITH GLASS NO. 9

(Shades on outdoor side, W/S = 1.2)

can lead to a large error in the final determined quantity when the heat flow rates are low. Undoubtedly some of the differences for a given shade are real ones and are the result of different radiation intensity distributions. Calculations based upon the standard brightness distributions of the *Illuminating Engineering Society*[7] showed that the diffuse transmittance value should vary with azimuth for a given solar altitude and that for zero azimuth, the value should decrease as the solar altitude increases (for vertical glass and zero azimuth, solar altitude and profile angle are the same).

Transmittance of Direct Radiation: Experimentally determined values of direct radiation transmittance are shown in Figs. 5, 6, 7, and 8 as functions of profile or shadow line angle. Values for white slats set at 30 and at 45 deg with a spacing ratio of 1.2 are shown in Fig. 5. The values for the 30 deg setting will be discussed in some detail. Most values shown were obtained by means of the condition A-2 tests (mainly with zero azimuth) but also included are values from one of the condition A-3 tests (fixed orientation). All data are for the

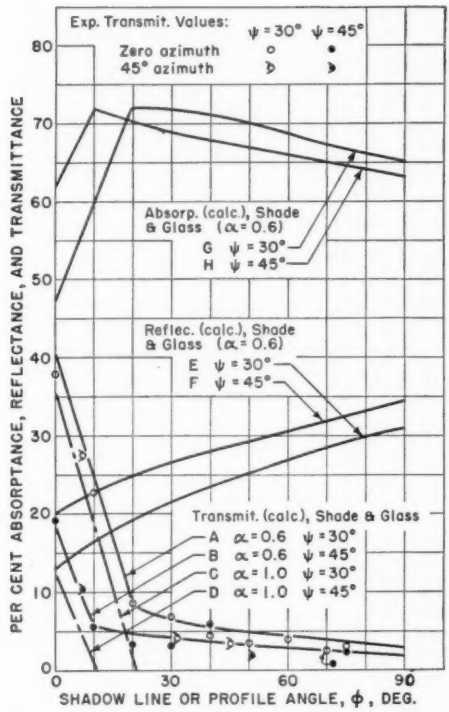


FIG. 6. ABSORPTANCE, REFLECTANCE, AND TRANSMITTANCE OF GREEN SLAT-TYPE SUN SHADES IN COMBINATION WITH GLASS NO. 9

(Shades on outdoor side, $W/S = 1.2$)

transmittance of the shade combined with glass No. 9, with the sunshade on the outdoor side. Columns 2 and 3 of Table A-1 in the Appendix give typical test data and illustrate the computations. The experimentally determined value of 0.28 for diffuse solar radiation was adjusted approximately for profile angle in accordance with the calculations just discussed.

Since the total solar energy absorptance value, α, is not known with exactness, theoretical curves A and B (Fig. 5), have been drawn for diffuse reflecting slats which have absorptances of 0.3 and 0.4 respectively. The data are seen to be in good agreement with the curve for $\alpha = 0.4$ but lower than the curve for $\alpha = 0.3$. It will be noted that the two test points for 20 deg profile angle are much below the theoretical curves. These two tests, as well as one test for 60 deg profile angle, were made on days when the diffuse radiation, measured by shielding the pyrheliometer from the direct rays of the sun, was very high in proportion to the direct radiation. Under such conditions much of the diffuse

radiation is concentrated close to the sun, and if part of it had been included with the direct radiation, the values would have been higher.

If the slat surfaces were to reflect 90 percent of the incident radiation diffusely and 10 percent specularly, the theoretical transmittance values would be 0.46, 0.31, and 0.17 for profile angles of zero, 10 and 20 deg respectively and $\alpha = 0.4$. For larger angles the values would be a little lower than curve B, Fig. 5. On the whole, curve B makes the best fit with the test data. The differences between curves A and B are not very significant from the standpoint of transmittance because the values are low in the range of greatest application, namely, profile angles greater than 20 deg.

Curve D, Fig. 5, is the transmittance for shades which absorb all radiation which strikes the slats and is the product of the glass transmittance and the opening ratio expressed as a percentage. The opening ratio is defined as the width of that portion of the opening between the slats through which a direct beam can pass without striking either slat surface, divided by the slat spacing. The difference between curves B and D for any given profile angle is the fraction of the incident radiation which is transmitted by way of reflections from the slat surfaces. Curve D shows that no direct beam will enter the room if the profile angle is 21 deg or more. Curve F is the theoretical curve for the shade alone for $\alpha = 0.4$.

Curves C and E (Fig. 5) are for white slats set at a 45 deg angle, with perfectly diffusing slat surfaces, and $\alpha = 0.4$ and 1.0 respectively. The test points, based upon diffuse transmittance values adjusted as before, though a little low, are in reasonably good agreement with the theoretical curve. Note that the cut-off point for $\psi = 45$ deg is at approximately 11 deg.

Also shown in Fig. 5 are the calculated curves of reflectance and absorptance for the shades in combination with the glass. All curves have finite values out to $\phi = 90$ deg at which point transmittance and absorptance values become zero and the reflectance value becomes 100 percent. Because of the low transmittance values, and also low glass absorptance (2 percent), the reflectance and absorptance curves are substantially the same for both shades alone and shades plus glass.

Note that the 45 deg slat setting gives substantially lower absorptances and higher reflectances than the 30 deg setting for profile angles greater than the cut-off angle. The transmitted gain will be small in either case, but since the 30 deg setting will admit somewhat more light than the 45 deg setting, the 30 deg setting might be preferred for some orientations.

The slight curvature of the slats was found by investigation to have no significant effect on transmittance.

Fig. 6 gives similar data for the shade with green slats set at 30 and 45 deg. Theoretical curves A and B have been drawn for perfectly diffusing reflecting slats with $\alpha = 0.60$. The test points are in reasonably good agreement with these curves. In view of the experimental error involved, particularly at angles greater than 30 deg, the points would fit the theoretical curves for $\alpha = 0.55$ nearly as well. These would be one or two percentage points higher. The figure also shows calculated curves of absorptance and reflectance.

The experimentally determined transmittances for the specular aluminum slats set at 30 deg with a spacing ratio of 1.2 in combination with and outside of glass No. 2 are shown in Fig. 7. The zero azimuth condition A-2 tests are in good agreement with curve B, which is the theoretical curve for specular reflect-

NOMENCLATURE

α = solar energy absorptance, dimensionless.
β = altitude angle, (see Fig. 1) degrees.
γ = azimuth angle, (see Fig. 1) degrees.
θ = incident angle, (see Fig. 1) degrees.
ϕ = shadow line or profile angle, (see Fig. 1 and Table 1) degrees.
ψ = slat angle, (see Table 1) degrees.
A_d = absorptance of shade alone for diffuse solar radiation, dimensionless.
$A_{d,g,s}$ = absorptance of glass and shade, with shade inside for diffuse solar radiation, dimensionless.
$A_{d,s,g}$ = absorptance of shade and glass with shade outside for diffuse solar radiation, dimensionless.
D = slat thickness, (see Fig. 1) inches.
S = slat spacing, (see Fig. 1) inches.
T_D = transmittance of shade alone for direct solar radiation, dimensionless.
$T_{d,s}$ = transmittance of shade alone for diffuse solar radiation, dimensionless.
$T_{d,s,g}$ = transmittance of shade and glass, with shade outside, for diffuse solar radiation, dimensionless.
W = slat width, (see Table 1) inches.

ing slats with $\alpha = 0.16$. In accordance with theory that $T_D =$ zero when $\psi + \phi$ is greater than 60 deg profile angle, the curve goes to zero at 60 deg. However, the test points for ϕ greater than 60 deg show small positive values. These represent differences of one to two Btu per (hr) (sq ft) between the correct values, zero, and the experimental values computed with the experimental value for diffuse radiation transmittance adjusted for profile angle. Had a constant value of 0.49 for diffuse transmittance been used, the values would have been negative by about the same amounts.

It will be noted that Curve C (Fig. 7), transmittance for shades inside the glass, does not go to zero at 60 deg. This is so because the radiation reflected from the slats strikes the glass with such a high incident angle that a sizeable fraction is reflected. These rays return to the slat system with a negative profile angle for which the transmittance of this shade is about 0.90. The slat thickness spacing ratio, 0.025, is sufficient to put the theoretical curves drawn about two percentage points below the theoretical values for a shade which has zero thickness ratio.

It will be noted also that the calculated reflectance curve H (Fig. 7) is very low at high angles and then rises abruptly to a constant value of 84 percent at $\phi = 60$ deg. Curve G for the shade indoors is influenced by the glass reflectance. The absorptance curves and data will be discussed in a later section.

Fig. 8 shows values for shade D outside of and in combination with glass No. 2. A theoretical curve, Curve B, for diffuse reflecting slats with $\alpha = 0.8$ is in good agreement with the experimental data. As before a value of diffuse transmittance which varied with profile angle was used in the computations, which are illustrated by columns 2 and 3 of Table A-1. Comparison of Curve D for $\alpha = 1.0$ with Curve B shows that very little radiation passes through the

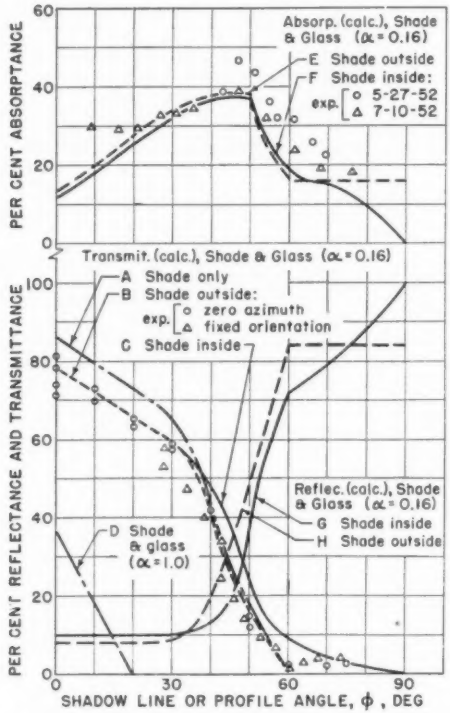

Absorp. (calc.), Shade & Glass (α = 0.16)
E Shade outside
F Shade inside:
exp. o 5-27-52
exp. Δ 7-10-52
PER CENT ABSORPTANCE
Transmit. (calc.), Shade & Glass (α = 0.16)
A Shade only
B Shade outside:
exp. o zero azimuth
exp. Δ fixed orientation
C Shade inside
Reflec. (calc.), Shade & Glass (α = 0.16)
G Shade inside
H Shade outside
D Shade & glass (α = 1.0)
PER CENT REFLECTANCE AND TRANSMITTANCE
SHADOW LINE OR PROFILE ANGLE, φ, DEG

FIG. 7. ABSORPTANCE, REFLECTANCE, AND TRANSMITTANCE OF SPECULAR ALUMINUM SLAT-TYPE SUN SHADE C IN COMBINATION WITH GLASS NO. 2

(W/S = 1.2, D/S = 0.025, W = 2.0 in., ψ = 30 deg)

shade by reflections from the slats. Most of this is due to the first reflection from the upper side of the slats through the rather wide spacing. The slat thickness spacing ratio was taken into consideration in computing the theoretical curves. Curve D shows that the cut off point is at approximately 42 deg profile angle for this shade. The theoretical transmittance curve, C, for slats inside the glass is identical with Curve B except for the dotted extension between 70 and 90 deg profile angle.

Fig. 8 also gives calculated reflectance and absorptance curves for the shade in combination with glass No. 2. The absorptance curves are discussed in the next section.

Absorptance of Solar Radiation: Absorptance values were determined from the data obtained in the tests for condition B, *i.e.*, with sunshade on the indoor

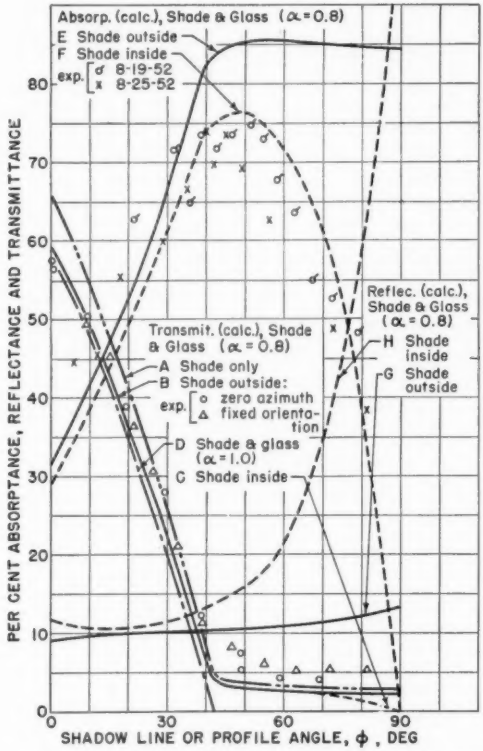


FIG. 8. ABSORPTANCE, REFLECTANCE, AND TRANSMITTANCE OF DARK SLAT-TYPE SUN SHADE B IN COMBINATION WITH GLASS NO. 2

(W/S = 0.8, D/S = 0.085, W = 0.047 in., ψ = 20 deg)

side of the glass. Only in this arrangement could the heat balance equation which follows be evaluated:

$$\begin{Bmatrix}\text{Fraction absorbed} \times \text{incident solar radiation}\end{Bmatrix} = \begin{Bmatrix}\text{Convection and radiation gain from shade and glass}\end{Bmatrix} + \begin{Bmatrix}\text{Convection heat loss, glass to outdoor air}\end{Bmatrix} + \begin{Bmatrix}\text{Low temperature radiation loss, glass to outdoor environment}\end{Bmatrix} + \begin{Bmatrix}\text{Rate of heat storage in glass and shade}\end{Bmatrix} \quad (1)$$

Attempts to determine the absorptance of diffuse solar radiation were not altogether satisfactory because of difficulty in accurately evaluating the various terms of the heat balance equation. Inexactness in the selection of convection conductances and diffuse solar radiation transmittances appropriate to the test conditions as well as in the measurements of incident low temperature radiation.

led to such wide variations in the determined value that an average value could be no more than a rough *order of magnitude* value. For example, an error of 5 Btu per (hr) (sq ft) in the determination of the incident low temperature radiation, which is an error of 5 percent in the case of the example given in column 5 of Table A-1 (see item 8) will produce an error of more than 20 percent in the value of diffuse absorptance.

Therefore, only calculated values, $A_{d,g,s}$ of the fraction of the incident diffuse solar radiation absorbed by the glass and shade together with the shade indoors, are given in Table 4. These were computed for an above-the-horizon source having a uniform intensity distribution. Because the experimentally determined transmittances were found to be higher than calculated, the absorptance values in practical cases should be somewhat lower than the table values. Other calculated values are for shades alone, and shades plus glass with the shade outside. When glass such as No. 2 or No. 9 is on the outdoor side of the shade, it absorbs about

TABLE 4—CALCULATED VALUES FOR DIFFUSE SOLAR RADIATION ABSORPTANCE FOR SLAT-TYPE SUN SHADES

SHADE	COLOR	SLAT WIDTH TO SPACING RATIO, W/S	SLAT ANGLE ϕ	ABSORPTANCE		
				Shade & Glass Shade Outside $A_{d,g,s}$	Shade & Glass Shade Inside $A_{d,s,g}$	Shade Alone A_d
A	White	1.2	30	0.44	0.51	0.49
A	White	1.2	45	0.43	0.48	0.47
B	Green	1.2	30	0.57	0.67	0.65
B	Green	1.2	45	0.60	0.69	0.68
C	Specular aluminum	1.2	30	0.23	0.28	0.25
D	Treated bronze	0.8	20	0.56	0.68	0.64

2 percent of the incident radiation. When the glass is on the indoor side, it absorbs a small fraction, approximately 0.02 times the shade transmittance.

Total solar absorptance values for white and green shades with slats set at 30 deg are shown in Fig. 9 as a function of profile angle. The theoretical curves drawn are for $\alpha = 0.4$ and $\alpha = 0.6$ respectively and are for glass No. 2 and the sun shade together, with the latter inside. It should be noted that these curves are for direct radiation only, whereas in computing the experimental values from the heat balance data, direct and diffuse radiation were lumped together. This accounts for part of the scatter, since the absorptances of the two components are not the same throughout the range of profile angles. However, the greater part of the scatter is due to the necessarily inexact evaluation of the heat balance terms as previously described.

Typical data and computations are listed in column 4 of Table A-1. On the whole the agreement of the data with the appropriate curves is reasonably good. The absorptance of the shades only, when they are inside the glass, would be represented by curves about three percentage points lower.

Fig. 9 also gives calculated reflectance and transmittance curves for the shade-glass combinations when the shade is on the indoor side. The principal differences between these curves and those for shades outside, which were given in Figs. 5 and 6, are between 70 and 90 deg profile angle.

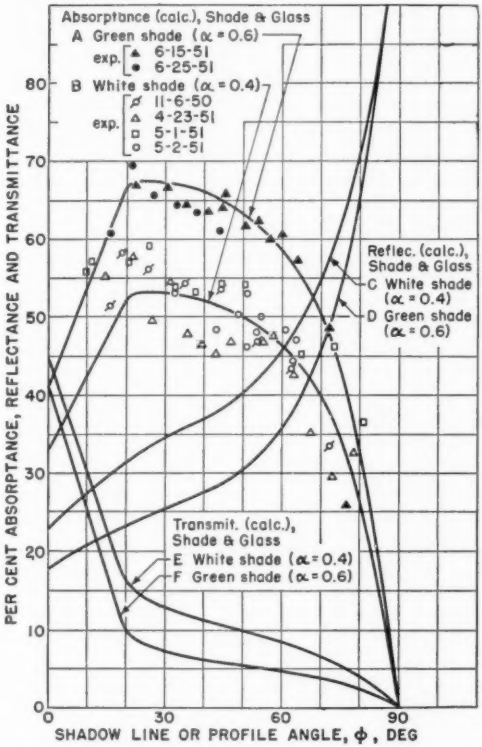


FIG. 9. ABSORPTANCE, REFLECTANCE, AND TRANSMITTANCE OF WHITE SHADE A AND GREEN SHADE B IN COMBINATION WITH GLASS NO. 2

(Shades on indoor side, $W/S = 1.2$, $W = 2.0$ in., $\psi = 30$ deg)

Total solar energy absorptance values for the specular reflecting aluminum slats in combination with glass No. 2 are compared in Fig. 7 with the theoretical values for direct radiation represented by curve F. Absorptance values for slats alone inside the glass would be about two percentage points lower. Curve E gives calculated values for shade and glass together with the shade outside.

Fig. 8 compares experimentally determined absorptance values for shade D inside glass No. 2 with the theoretical curve F. Curve E gives calculated values for the shade outside the glass.

DISCUSSION OF RESULTS

The adequacy of the mathematical analysis of transmittance and absorptance of solar radiation by slat-type sun shades must be judged principally on the agree-

ment, or lack of it, between the theoretical and the experimentally determined values of transmittance of direct solar radiation, with due regard to the limitations of our experimental treatment of such a complex problem. The determination of absorptance values is in part dependent upon the experimentally determined transmittance values, so that the absorptance values are not a wholly adequate basis of judgment.

References to Figs. 5, 6, 7, and 8 illustrate the fact that the direct solar radiation transmitted by a slat-type shade can be broken down into two components. One is what might be termed the *straight-through component;* the other is the *reflected-through component.* The mathematical analysis previously presented showed that the first component is a simple trigonometric function of the slat geometry and profile angle. The second is dependent not only upon slat geometry and profile angle but also upon the reflectance or absorptance of the slat surface.

Fig. 6 for green slats and Fig. 8 for the small dark slats, shade D, demonstrate good overall agreement between the theory and experiment for shades with diffusely reflecting slats, although the absorptance values for the total solar spectrum are not known with exactness. However, the values are so high that differences of as much as 0.05 between the true values and the nominal values of 0.6 and 0.8 used in computing the theoretical curves, will not result in experimentally measurable differences in transmittance. Hence, the theory connected with the reflected-through component is not thoroughly tested in spite of the good overall agreement. In the absorptance range of 0.3 to 0.4 the experimental method seems to be adequate to test the theory of the reflected-through component. Unfortunately present knowledge of the solar absorptance value of the white slat, illustrated by Table 2 and the accompanying discussion, is inadequate. However, differences between values for the green slats and the white slats, for the same shade geometry and profile angle, are in good agreement with the differences to be expected theoretically for absorptance values of 0.6 and 0.4. The maximum effect of α occurs at the cut-off point. The difference is 7 percentage points at 20 deg profile angle for $\psi = 30$ deg and about 5 points at $\phi = 10$ deg for $\psi = 45$ deg. The differences between transmittance values for slat surface absorptance of 0.6 and 0.3 would be 11 and 8 percentage points for $\psi = 30$ deg and $\psi = 45$ deg respectively. Judged on this basis, the theory related to the reflected-through component would appear to have practical usefulness.

The theory for specular reflecting slat surfaces is quite straightforward, though the equations appear rather formidable. The absorptance of 0.16 for specular aluminum is low enough that both the straight-through and the reflected-through components are adequately tested and verified by experiment. A change of 0.05 in α in the region of 0.15 will produce a change of almost 10 percentage points at $\phi = 30$ deg.

The effect of energy distribution on shade performance must not be ignored. Calculations based on Moon's[4] suggested standards for direct solar radiation for the white and aluminum slats show that the change in energy distribution with air mass causes no significant change in the absorptance for the total spectrum. This may not hold for the visible region and therefore should be considered in illumination problems related to sunshades.

The effect of heat absorbing glass on the behavior of an indoor shade should be significant. Inspection of Fig. 3 shows that solar energy which is depleted

in the infra-red region will raise the average absorptance of specular aluminum slats and decrease it for the white slats. Conversely, a shade on the outdoor side will influence the performance of the heat absorbing glass.

The effects of such factors as polarization and change of absorptance with change of incident angle were not treated in the theory, though the second can be treated; nor are they disclosed in the experimental work.

On the whole the theoretical treatment of slat-type sunshades seems to be confirmed by these experiments as a practical working tool in the development of design data for shades. The specification of design conditions as to shade reflection characteristics and geometry is an important practical problem but outside the scope of this paper. The very important problem of convection and radiation gain from shade-glass combinations must be solved in order to complete the basic knowledge necessary in developing sound design data. This will be treated in a paper now in preparation.

ACKNOWLEDGMENT

The authors acknowledge with thanks the assistance of their colleagues of the A.S.H.V.E. Research Staff in the work described in this report and are grateful for the interest and technical guidance of the members of the TAC on Heat Flow Through Glass. In particular they thank W. B. Ewing of the Ewing Development Co. and J. R. Young of the Kaiser Aluminum Co. for technical data furnished.

REFERENCES

1. A.S.H.V.E. RESEARCH REPORT No. 1460—The Shading of Sunlit Glass: An Analysis of the Effect of Uniformly Spaced Flat Opaque Slats, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 377).

2. A.S.H.V.E. RESEARCH REPORT No. 1348—Solar and Total Heat Gain Through Double Flat Glass, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 407).

3. A.S.H.V.E. RESEARCH REPORT No. 1333—Measurements of Solar Heat Transmission Through Flat Glass, by G. V. Parmelee, W. W. Aubele, and R. G. Huebscher (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 165).

4. Proposed Standard Solar Radiation Curves for Engineering Use, by P. Moon (*Journal of the Franklin Institute*, Vol. 230, No. 5, November 1940, p. 583).

5. Private communication from R. V. Dunkle and J. T. Gier (*Institute of Engineering Research*, University of California, Berkeley, California, June 1951).

6. Some Reflection and Radiation Characteristics of Aluminum, by C. S. Taylor and J. O. Edwards (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 179).

7. Recommended Practice of Daylighting, by the Daylighting Committee of the *Illuminating Engineering Society* (*Illuminating Engineering*, Vol. 45, No. 2, February 1950, p. 129).

APPENDIX A

TABLE A-1—TYPICAL TEST DATA AND CALCULATIONS BY WHICH ABSORPTANCE AND TRANSMITTANCE VALUES WERE OBTAINED

All Values are for White Shade with W/S = 1.2 and ϕ = 30 deg in Combination with Glass No. 9.

TEST NO.	1	2	3	4	5
DATE	8-15-50	8-16-50	8-16-50	4-23-51	4-23-51
TIME (Sun)	12:30-1:00	2:00-2:30	10:45-11:15	4:30	9:30-10:00
OBSERVED VALUES					
1. Location of sunshade	Out.	Out.	Out.	In.	In.
2. Incident angle	—	0	50	46.6	—
3. Tilt of calorimeter, deg from vertical	0	49.2	8.3	0	0
4. Azimuth of calorimeter, deg west from south	180	57.5	331.3	45	45
5. Profile angle, deg	—	0	50.0	31.8	—
Incident radiation, Btu/(hr) (sq ft):					
6. Direct solar	—	251.5	148.0	142.8	—
7. Diffuse solar	25.0	43.8	49.3	32.4	22.0
8. Low temperature	—	—	—	108.3	95.0
Wind Conditions:					
9. Wind velocity, mph	4.8	7.0	5.4	4.2	6.6
10. Wind direction	N	W	NW	NW	NW
Temperatures, Fahrenheit:					
11. Calorimeter, t_c	76.6	80.6	79.4	74.8	69.5
12. Glass, t_g	81.6	90.5	88.2	73.8	51.0
13. Sun shade, t_s	82.1	94.3	99.8	97.5	66.2
14. Outdoor air, t_o	83.3	86.5	85.1	52.4	45.6
DERIVED QUANTITIES COMPUTED FROM OBSERVED VALUES All Heat Flows in Btu per (hr) (sq ft)					
Calorimetry:					
15. Gain by coolant	17.6	134.0	34.2	65.7	−29.2
16. Calorimeter heat losses	−5.3	−5.1	−1.7	7.4	18.4
17. Corrected gain[a]	12.3	128.9	32.5	73.1	−10.8
Derived Quantities, Shade Outdoors:					
18. Gain due to convection and radiation from glass[b]	5.5	11.3	10.2	—	—
19. Gain due to transmitted solar radiation[c]	6.8	117.6	22.3	—	—
20. Fraction of diffuse solar radiation transmitted	0.27[d]	0.32[e]	0.24[e]	—	—
21. Gain due to transmitted solar radiation[f]	—	103.6	10.5	—	—
22. Fraction of direct solar radiation transmitted[g]	—	0.41	0.07	—	—
Derived Quantities, Shade Indoors:					
23. Fraction of direct solar radiation transmitted[h]	—	—	—	0.12	—
24. Fraction of diffuse radiation transmitted[e]	—	—	—	0.28	0.28
25. Gain due to total transmitted solar radiation[i]	—	—	—	26.2	6.2
26. Gain due to convection and radiation from shade and glass[k]	—	—	—	44.3	−17.0
27. Convection loss, glass to outdoor air[m]	—	—	—	22.4	5.1
28. Radiation loss, glass to outdoors[n]	—	—	—	29.8	21.6
29. Rate of heat storage, glass and shade	—	—	—	−2.2	1.6
30. Total solar radiation absorbed by glass and shade[p]	—	—	—	94.3	11.3
31. Fraction of total solar radiation absorbed[q]	—	—	—	0.54	0.52

[a] Item 15 + item 16.
[b] Based upon item 12 − item 11.
[c] Item 17 − item 18.
[d] Item 19 ÷ item 7.
[e] Values from Table 3 adjusted for variation of $T_{d.s.g}$ with ϕ.
[f] Item 19 − (item 20 × item 7).
[g] Item 21 ÷ item 6.
[h] Values from curve B, Fig. 5.
[i] (Item 23 × item 6) + (item 24 × item 7)
[k] Item 17 − item 25, with allowance for partial shading by window frame.
[m] Based upon items 9, 10, 12 and 14.
[n] Based upon items 8 and 12.
[p] Items 26 + 27 + 28 + 29.
[q] Item 30 ÷ (item 6 + item 7).

DISCUSSION

R. W. McKINLEY, Pittsburgh, Pa.: Referring to the diagram in Table 1, it will be noted that the paper deals with direct radiation from the sun. It is pointed out in the paper that the slat type shade is adjusted in various positions in order to exclude direct radiation transmission to some degree.

A factor which has been disregarded in this paper is the reflected solar radiation from a horizontal surface such as a flat solid surface or the ground. The reflected radiation from such a surface may pass between these slats when the slats are adjusted to exclude direct solar radiation. The magnitude of this reflected component is worthy of consideration and possible study before the authors complete the investigation.

As a practical matter, adjustment of the slats is a compromise which involves at the same time the desire for maximum light transmission into the room, maximum natural ventilation, and a minimum of heat transmission into the room.

It would be most interesting if the authors could indicate the performance of the shade as affected by light transmission (which has been studied by some other researchers), ventilation value, and the shading effect for both direct and reflected radiation, so that a practical design compromise could be made for slat type shades.

CYRIL TASKER, Cleveland, Ohio: Those of you who have seen papers by Mr. Parmelee over the last several years realize that he has been carrying on a very interesting and a long-range study.

All the first studies made under the Technical Advisory Committee were on unshaded glass, but even when these were being made some people were asking us to obtain data for some shades to show how much solar heat is excluded by shades. It was obvious that we could not tell what shades did until we knew something about what happened without shades. A rather extensive program was therefore carried on over the years, and a number of papers were presented before the Society on the performance of unshaded glass. This is the second on the effects of shades.

The end point of our testing is to obtain design data on shaded sunlit glass to complement the data on unshaded glass which is now in THE GUIDE.

This work has attracted much attention not only in this country but abroad, and as a result, Mr. Parmelee has become quite well known as an expert on the subject both here and outside of this country. As a result, we recently had a tentative request for the loan of his services in the Middle East to assist governments on certain building problems.

There has been a renewal of interest in this work on the part of the *Illuminating Engineering Society* following attempts to bring to the attention of the two Societies the many interests they have in common for cooperative work. When we started the work on heat transfer through glass, it did not seem likely that the shading part of the work was going to be the thing that ultimately brought together the *Illuminating Engineering Society* and the A.S.H.V.E. but that seems to be the present trend.

We have felt that the shading of sunlit glass was a problem which should be handled by other persons than those having a primary interest in glass. I think that some of the people who are interested in shading are missing a real opportunity in not participating themselves, while the glass people go ahead trying to get the work done.

It is interesting to note that many of our Society activities are related to the activities of some other group. If we can only bring the right people together we can get cooperative action from unexpected sources.

R. A. MILLER, Pittsburgh, Pa.: I have been a member of this Society for 23 years. I have been a member of the *Illuminating Engineering Society* for about the same length of time. I have done everything I could to get the two Societies to recognize the fact that they had many interests in common and to do some cooperative work. When we first started research work on heat transfer through glass, no one had any

idea that the shading of glass would be the thing that ultimately brought the *I.E.S.* into the picture; or brought the A.S.H.V.E. into the *I.E.S.* picture; but that seems to be the present trend.

We have felt that the shading of sunlit glass was a problem which should be handled by people other than those in the glass industry. Some of these results seem to indicate that that probably isn't true, and that those interests concerned with shading are losing an opportunity by not participating, while the glass people are trying to get the work done.

It is interesting to note that each of our Society activities ties into the activity of some other group and, therefore, if we can only get the right people together, we can obtain cooperative action from unexpected sources. Mr. Cotton's presentation of central power companies' interest is exactly in line with these thoughts, and is concerned with these same problems of obtaining cooperation.

AUTHORS' CLOSURE (Mr. Parmelee): Discussion in the meeting of the TAC on Heat Flow Through Glass seemed to indicate that research on shading was reasonably complete, but after hearing Mr. McKinley's remarks, I am sure this is not true.

The problem of selecting the proper slat angle, color, and spacing is a real one for the designer. The purpose of windows is to admit natural light and, from the standpoint of cooling loads, there is some evidence to indicate that natural light is more economical in many cases than artificial light. As in all design problems, there has to be some compromise between natural lighting, minimum solar heat gain, and, in non-air conditioned spaces, a third factor, ventilation.

The A.S.H.V.E. has no study of natural ventilation in progress. However, the Engineering Experiment Station of Texas Agricultural and Mechanical College has carried on some wind tunnel studies of ventilation through openings covered by slats. Perhaps out of this work the engineer can find enough information to assist him in solving his problems.

The matter of ground reflected solar radiation was an important one experimentally. It was a variable factor and something we could not do much about except to include its effect in the overall performance of a given shade. This subject has been discussed in *I.E.S.* meetings, because it is important in the illumination of classrooms and other buildings.

Concrete parking lots, light colored gravel, and similar surfaces reflect a lot of solar radiation in an upward direction for which the slat-type shades have a high transmittance. It is conceivable that under some conditions it might be desirable to set the slats of a shade in a direction opposite to the usual one.

I cannot give, at present, a good answer to Mr. McKinley's request for an overall picture of sunshade performance in terms of all these factors previously brought out, but I believe that with the information we have developed and with that of others, we can now begin to find the answers.

1475

No. 1475

MAXIMUM DOWNWARD TRAVEL OF HEATED JETS FROM STANDARD LONG RADIUS *A.S.M.E.* NOZZLES

By LINN HELANDER*, SHEE-MANG YEN**, AND R. E. CRANK**,
MANHATTAN, KAN.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *Industrial Unit Heater Association* and the Engineering Experiment Station of Kansas State College.

GRAPHS AND FORMULAS, suitable for estimating the maximum downward travel of perpendicular jets of heated air from long radius *A.S.M.E.* nozzles and having practical guidance value for air conditioning engineers and designers of unit heaters, are presented in this paper. The graphs, the formulas, and the data upon which they are based are the results of research conducted at Kansas State College under the guidance of the Technical Advisory Committee on Air Distribution†. A previous paper[1] on this research, presented before the Society in 1948, gave data for jets of heated air projected vertically downward from (1) a straight run of pipe 6½ in. in diameter, and (2) a convergent nozzle with an orifice diameter of 5.75 in. Since 1948, studies have been made of heated jets projected vertically downward from long radius *A.S.M.E.* nozzles with orifices ranging from 2 to 17 in. in diameter. This paper deals with the maximum downward travel, or throw, of these jets. The range of orifice velocities and the range of orifice temperatures employed in the studies are given in Table 1.

TEST PROCEDURES AND EQUIPMENT

Velocities were measured by means of a pitot tube coupled to either a sensitive analytical balance, modified to measure differentials of pressure of the order of 1/5 in. of atmospheric air (equivalent to 0.0002 in. water), or to a

* Professor and head, Department of Mechanical Engineering, Kansas State College. Member of A.S.H.V.E.

** Assistant Professor, Department of Mechanical Engineering, Kansas State College.

† Personnel: G. B. Priester, *chairman;* W. A. Grant, *vice chairman;* J. M. Black, H. F. Brinen, R. M. Conner, S. H. Downs, S. F. Gilman, Linn Helander, F. B. Holgate, W. O. Huebner, W. W. Kennedy, E. J. Kurek, R. D. Madison, G. E. McElroy, L. G. Miller, H. C. Pierce, C. H. Randolph, T. H. Troller, R. D. Tutt, G. L. Tuve.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

micromanometer which could be read to 0.001 in. water. Temperatures were measured by means of shielded thermocouples made of standard thermocouple wire. These were connected to an electronic precision indicator. To locate the bottom of the jet, a pitot tube connected to the modified balance and a thermocouple connected to the indicator were simultaneously moved upward into the jet from a point well below the bottom of the jet. The level of the bottom of the jet was denoted as that level at which the first sustained evidence of velocity was observed, together with a marked change in the temperature gradient and marked fluctuations in temperature. Occasionally, smoke was used as a supplementary means for locating the bottom of the jet.

The thermocouple for measuring the room-air temperature, T_a, was located at the instrument table 30 in. above the floor. Velocities at the orifice recorded

TABLE 1—RANGE OF TEST CONDITIONS

SIZE OF NOZZLE IN.	RANGE OF L_{max}/D_o	RANGE OF $(T_o—T_a)$ DEG F	RANGE OF U_o FT/SEC
	L_{max}/D_o smaller than 8		
4	1.5—6.8	70—90	2.5—5.5
	L_{max}/D_o between 8 and 45		
2	11—45	25—60	5—19
4	12—45	30—160	8—46
6	14—36	110—180	14—43
8	13—38	120—190	21—53
10	9—29	60—200	16—42
12	12—28	50—160	16—42
14	8—22	90—160	17—32
17	9.5—20	80—140	19—31
	L_{max}/D_o greater than 45		
2	46—92	30—65	15—46

as U_o, and temperatures recorded as T_o, were measured at the center of the orifice.

Fig. 1 shows the arrangement of test equipment in the laboratory. Fig. 2 shows dimensions and schematic arrangement of equipment for studying the throw of jets from nozzles having diameters of 6 in. or larger. Air from the blower passes through a heat exchanger to the plenum chamber, and from there to the nozzle. The plenum chamber is 48 in. wide, 50 in. deep, and 54 in. high. It is well insulated. The bottom of the plenum chamber is 34 ft above the level of the floor. The instrument table is shown in Fig. 2 in its approximate location. The nozzles used were spun aluminum nozzles. None of the nozzles except the 4-in. nozzle was insulated. The 4-in. nozzle was covered on the outside with an asbestos insulation about $\frac{1}{8}$ in. thick.

The volume of air delivered to the nozzle was controlled both by varying the speed of the direct current motor and by regulating a remotely controlled damper on the blower inlet. Minor variations in temperature and velocity were observed at the orifice.

All operations were controlled from the floor level. The thermocouple, pitot tube, and smoke gun, when used, were mounted on a motor operated instrument

FIG. 1. ARRANGEMENT OF TEST EQUIPMENT IN LABORATORY

trolley which traveled north and south on an instrument carriage. The carriage in turn was motor operated and traveled east and west on an instrument elevator. The elevator could be moved to any required level by means of a hand-operated winch.

For the jets from the 2-in. and 4-in. nozzles, the test set-up described in the previous paper[1] was employed. In this set-up, the base of the plenum chamber was 23 ft above the floor, and the platform that supported it extended into the

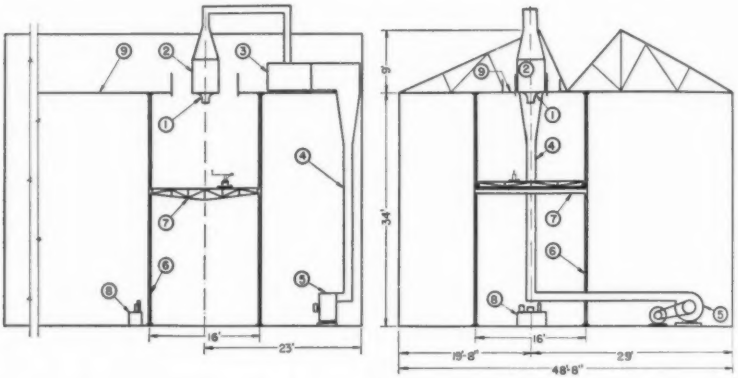


FIG. 2. DIMENSIONS AND SCHEMATIC ARRANGEMENT OF EQUIPMENT FOR TESTING LARGE NOZZLES

Reference numbers: (1) nozzle, (2) plenum chamber, (3) heat exchanger, (4) duct system, (5) blower, (6) supporting column, (7) instrument elevator, (8) instrument table, (9) catwalk

free space of the laboratory from upper-level flooring which formed a ceiling on one side of the jet.

DATA ON THROW

Fig. 3 is a composite plot of the data on the downward throw of perpendicular jets of heated air obtained since the publication of the previous paper[1]. Figs. 4 to 6 show these data plotted separately for each size of nozzle employed. The

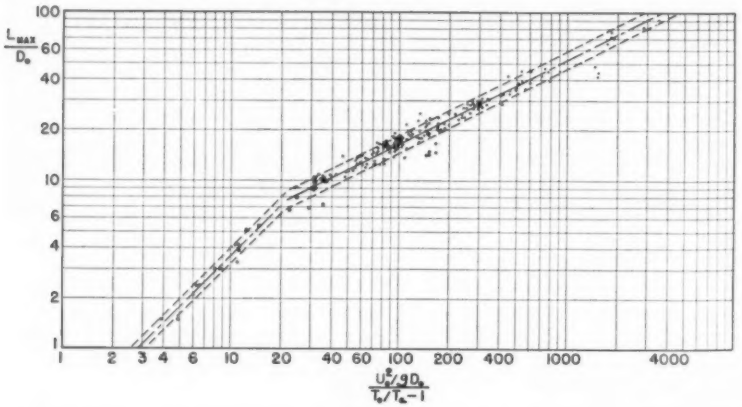


FIG. 3. THROW IN ORIFICE DIAMETERS, L_{max}/D_o, *vs.* BUOYANCY NUMBER, $(U_o^2/gD_o)/[(T_o/T_a) - 1]$

data plotted form a band, the boundaries of which lie 10 percent above and 10 percent below the curve of mean values. Within this band lie approximately two-thirds of all the plotted results.

Based on Fig. 3, the following equations for throw are obtained (Symbols are defined in list in Nomenclature):

1. For throws plotted in Fig. 3 equal to or greater than 8 orifice diameters, and for the range of conditions tabulated in Table 1,

$$\frac{L_{max}}{D_o} = 1.66\left(\frac{U_o^2/gD_o}{\frac{T_o}{T_a}-1}\right)^{\frac{1}{3}}, \pm 10\% \quad \ldots\ldots\ldots\ldots \quad (1)$$

2. For throws equal to or less than 8 orifice diameters, and for the range of conditions tabulated in Table 1,

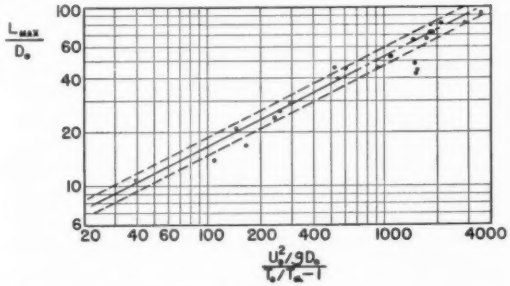


FIG. 4. THROW IN ORIFICE DIAMETERS, L_{max}/D_o, *vs.* BUOYANCY NUMBER, $(U_o^2/gD_o)/[(T_o/T_a) - 1]$, FOR JETS FROM 2-IN. NOZZLE

$$\frac{L_{max}}{D_o} = 0.36\left(\frac{U_o^2/gD_o}{\frac{T_o}{T_a}-1}\right), \pm 10\% \quad \ldots\ldots\ldots\ldots \quad (2)$$

INTERPRETATION AND DISCUSSION OF RESULTS

For throws between approximately 8 and 45 orifice diameters, Equation 1 and the corresponding band in Fig. 3 are supported by a substantial amount and range of test data. For throws in excess of 45 orifice diameters, this equation and band are supported by data on jets from a 2-in. nozzle. On the other hand, Equation 2 and its corresponding band are supported by data obtained from only one series of tests on jets from a 4-in. nozzle. The latter equation and its band therefore should be regarded simply as being indicative of the trend in the magnitudes of throws which are shorter than 8 orifice diameters.

The band drawn in Fig. 3 is a statistically established band within which the throw will probably lie. Figs. 4 to 6 show that the data for each size of jet fall principally within this band. The data for the jet from the 17-in. nozzle, how-

TABLE 2—RATIOS OF AVERAGE VELOCITY, $\overline{U}_o$, AVERAGE ELEVATION OF TEMPERATURE ABOVE ROOM TEMPERATURE, $\overline{T}_o - T_a$, AND AVERAGE BUOYANCY NUMBER, $\overline{B}_o$, TO CORRESPONDING CENTERLINE VALUES AT ORIFICE

1	2	3	4	5	6	7	8
DIAMETER OF ORIFICE IN.	NUMBER OF RUNS	CENTERLINE VELOCITY U_o FT PER SEC	CENTERLINE BUOYANCY NUMBER[a] B_o	$(\overline{U}_o/U_o)$ CALCULATED (FOOTNOTE b)	$(\overline{U}_o/U_o)$ MEASURED	$\frac{\overline{T}_o - T_a}{T_o - T_a}$ (FOOTNOTE c)	$\frac{\overline{B}_o/B_o}{(\text{COLUMN } 5)^2}$ (COLUMN 7)
17	2	16—22	40—71	0.98	0.98—0.99	0.99	0.96—0.97
10	5	13—34	24—83	0.97—0.98	0.98—0.99	0.99	0.95—0.96
6	6	21—40	129—518	0.97—0.98	0.98	0.98	0.96—0.98
4	5	9—28	58—1460	0.94—0.96	0.97	0.96—0.98	0.91—0.97
4	7	20—30	155—361	0.96—0.97	0.96	0.90—0.91	1.02—1.03
2	4	11—34	150—2390	0.94—0.96	0.93—0.96	0.96—0.98	0.92—0.94

[a] $B_o = (U^2_o/gD_o)/(T_o/T_a - 1)$; $\overline{B_o} = [(\overline{U}_o)^2/gD_o]/(\overline{T_o}/T_a - 1)$.

[b] Evaluated as being the same as the coefficient of discharge calculated from data on page 39 of *A.S.M.E.* publication PTC 19.5; 4-1949.

[c] Based on area-mean values of temperature at the orifice.

The values 0.90 to 0.91 in column 7 would have been increased probably 2 to 3 percent and the corresponding values in column 8 would have been reduced a like amount had the enthalpy-mean temperature been employed.

ever, lie mainly above the curve of mean values. The bottom of all jets fluctuated appreciably. The throw as plotted, therefore, is a time mean of the instantaneous values of a fluctuating throw.

The following four conditions under which the jets were studied are to be particularly noted:

1. Boundaries of the space into which the jets were projected were remote from the jets, so that the jets may be classified as approximately free jets.

2. Jets were free of whirl at the orifice and of artificially augmented turbulence.

3. The ratio of the cross-stream mean velocity at the orifice to the centerline velocity varied characteristically with the Reynolds number. For the conditions covered by Equation 1, this ratio had an approximate average value of 0.96 and the range of values shown in Table 2. Likewise, for the same conditions, the ratio of the mean elevation of temperature at the orifice to the centerline elevation, and the ratio of

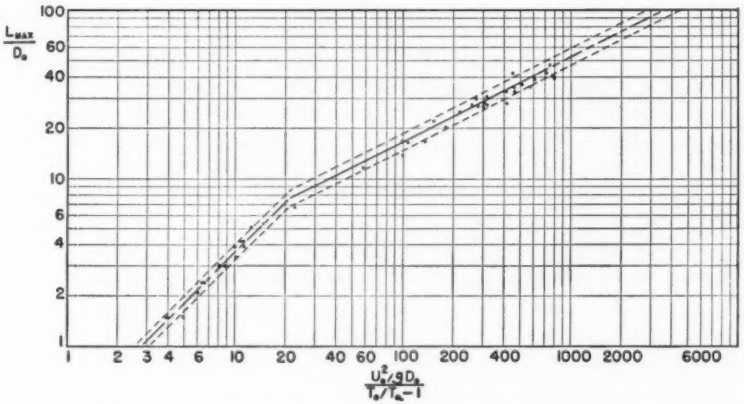


FIG. 5. THROW IN ORIFICE DIAMETERS, L_{max}/D_o, *vs.* BUOYANCY NUMBER, $(U_o^2/gD_o)/[(T_o/T_a) - 1]$, FOR JETS FROM 4-IN. NOZZLE

the mean buoyancy number at the orifice to the centerline buoyancy number (quantity in parenthesis in Equation 1) each had an average value of 0.96 and the range of values shown in Table 2. Equation 2 is based on low Reynolds numbers at the orifice and corresponding orifice-velocity profiles.

4. Nozzles employed were long radius nozzles suspended from the base of a plenum chamber and free on all sides.[2]

Conditions 1 to 4 must be approximated when Equations 1 and 2 are used to estimate the throw.

Studies are under way to determine the effect that orifice conditions, such as exist at the outlets of commercial types of unit heaters, would have on the throw.

QUALITATIVE DESCRIPTION OF JET

Fig. 7 shows the shape of a typical jet which has been made visible by means of smoke. Fig. 8 is a schematic representation of a jet typical of those studied

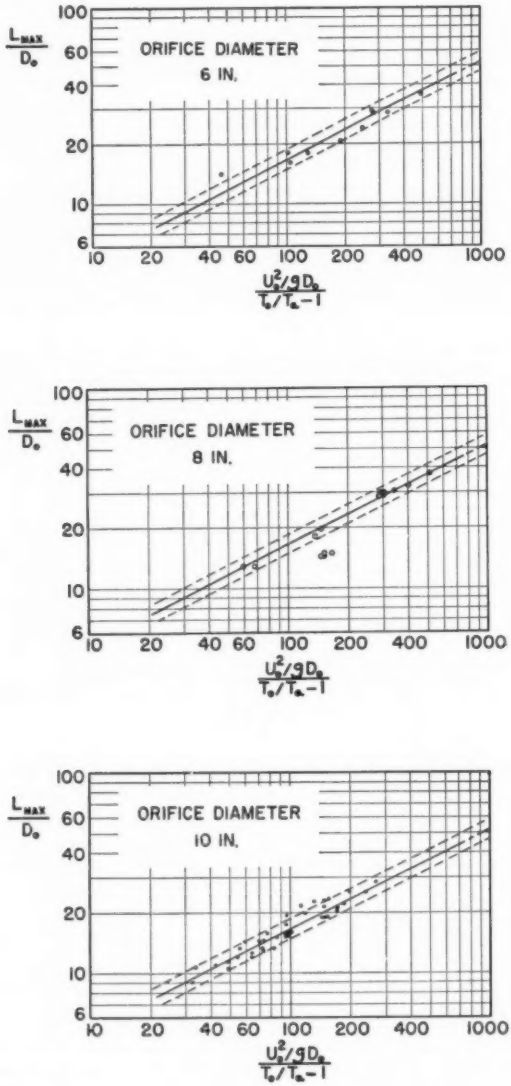


FIG. 6. THROW IN ORIFICE DIAMETERS, L_{max}/D_o, *vs.* BUOYANCY NUMBER, $(U_o^2/gD_o)/[(T_o/T_a)-1]$, FOR JETS FROM NOZZLES WITH ORIFICES 6 IN. TO 10 IN. IN DIAMETER

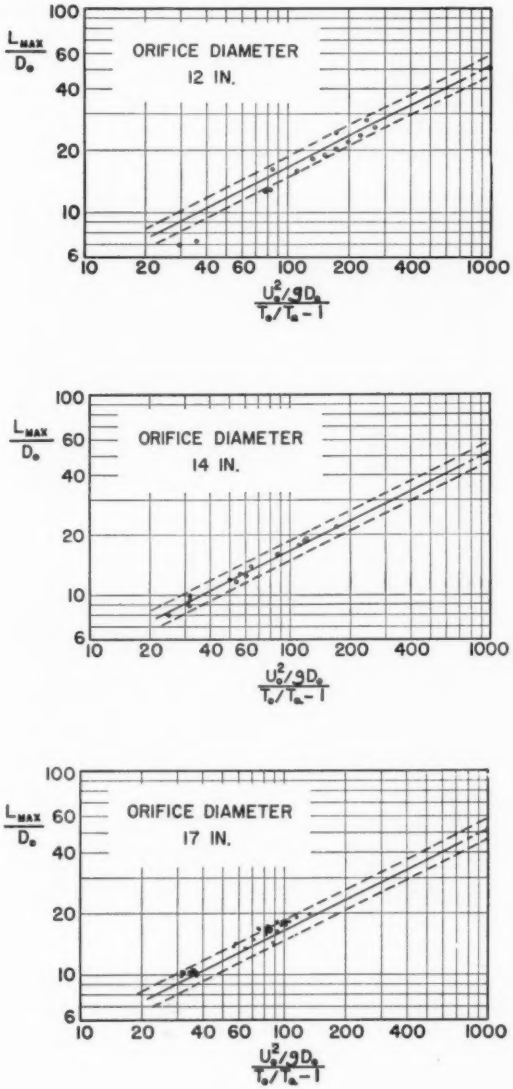


FIG. 6 *(concluded)*. THROW IN ORIFICE DIAMETERS, L_{max}/D_o, *vs.* BUOYANCY NUMBER, $(U_o^2/gD_o)/[(T_o/T_a)-1]$, FOR JETS FROM NOZZLES WITH ORIFICES 12 TO 17 IN. IN DIAMETER

where the throw was greater than approximately 18 orifice diameters. For convenience in dealing with the jet both analytically and descriptively, the jet is represented as having three zones or regions; (1) a primary zone, (2) a principal zone, and (3) a terminal zone.

More data on these three zones are needed before they can be definitively delineated. As represented in Fig. 8, the primary zone extends from the orifice to a level approximately 6 orifice diameters down from the orifice. At this level,

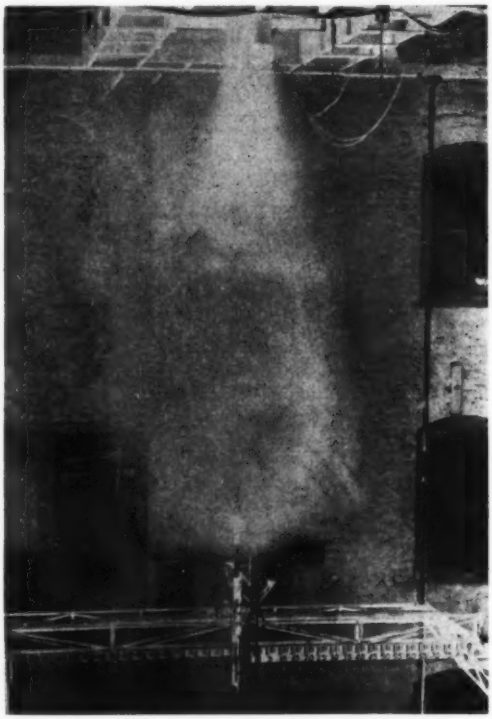


FIG. 7. JET MADE VISIBLE BY SMOKE

a point of inflection occurs in the curve of centerline velocities plotted at the right of the outline of the jet. This point has been taken as marking the nominal end of the primary zone. It has also been taken as the point at which the transition in flow from the pattern at the orifice to the pattern in the principal zone has nominally been completed. The principal zone may, for present purposes, be regarded as being that portion of the jet wherein: (1) Cross-stream temperature profiles may be presumed to form one family of profiles, the cross-stream velocity profiles another; and (2) Degree of consistency among the significant characteristics of the zone, including the curves of centerline velocity and tem-

NOMENCLATURE

a = a dimensionless entrainment factor equal to 2 tan (θ/2). See Fig. 9.

B_o = buoyancy number at orifice,

$$\frac{U_o^2/gD_o}{(T_o/T_a)-1}$$

$\overline{B_o}$ = Cross-stream average buoyancy number at orifice.

B_{ic} = buoyancy number at center of nominal interface between primary and principal zones,

$$\frac{U_{ic}^2/gD_i}{(T_{ic}/T_a)-1}$$

b_1 and b_2 = Dimensionless quantities to be determined experimentally.

C = A symbol in Equations 3 and 5 for dimensionless quantities to be determined experimentally.

c = A symbol used as subscript to indicate values at the vertical axis of the jet.

D_o = diameter of the jet at the orifice, feet.

D_i = diameter of the jet at the nominal interface between the primary zone and the principal zone, feet.

f = A symbol meaning *function of*. With a subscript, f denotes a specific form of function to be determined.

g = Acceleration of gravity, feet per (second) (second).

h_i = Distance from apex of principal zone, extended as in Fig. 9, to nominal interface between primary zone and principal zone, feet.

h_{max} = Distance from apex of principal zone, extended as in Fig. 9, to bottom of jet, feet.

i = subscript denoting value at nominal interface between primary zone and principal zone.

L_i = Distance from orifice to nominal interface between primary zone and principal zone, feet.

ΔL_{max} = distance from nominal interface between primary and principal zones to bottom of jet, feet.

L_{max} = distance from orifice to bottom of jet, feet.

o = subscript denoting value at orifice.

T_{ic} = temperature of jet at center of interface between primary zone and principal zone, Fahrenheit, absolute.

T_o = temperature of air at orifice, measured at center of orifice, Fahrenheit, absolute.

$\overline{T_o}$ = Cross-stream average temperature at orifice, Fahrenheit, absolute.

T_a = temperature of undisturbed room air, measured at instrument table approximately 30 in. above floor, Fahrenheit, absolute.

U_o = Velocity of air at orifice, measured at center of orifice, feet per second.

$\overline{U_o}$ = Cross-stream average velocity at orifice, feet per second.

U_{ic} = Local velocity of jet at center of interface between primary and principal zones, feet per second.

θ = Apex angle formed by tangents to boundary of principal zone at interface between primary zone and principal zone. See Fig. 9.

ρ_a = Density of room air at point T_a is measured, slugs per cubic foot.

ρ_{ic} = Density of jet air at center of interface between primary zone and principal zone, slugs per cubic foot.

ρ_o = Density of orifice air corresponding to temperature T_o, slugs per cubic foot.

Note: Nominal interface between primary zone and principal zone = Plane at which primary zone nominally ends and principal zone nominally begins. See Figs. 8 and 9.

perature, will be such that one is warranted in assuming that an analytical treatment of the zone will prove fruitful.

The principal zone extends from the end of the primary zone to the point in the jet at which the pattern of flow characteristic of the principal zone begins to decay progressively. The terminal zone extends from this point to the bottom of the jet. The terminal zone, at least the bottom portion of it, is a zone of marked instability.

Typical curves of velocity and temperature along the vertical axis of the jet are shown in Fig. 8 at the right of the jet; typical velocity and temperature profiles are shown at the left. The temperature profiles are flatter than the velocity profiles; that is, as the jet is traversed along a diameter, the percentage of change in the local jet temperature, measured relative to the established

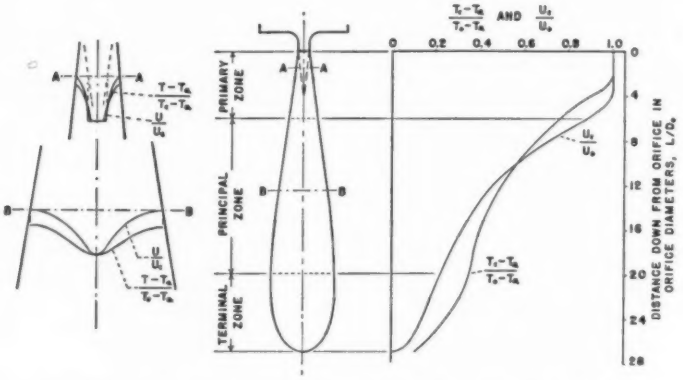


FIG. 8. SCHEMATIC REPRESENTATION OF A LONG THROW JET

Velocities and temperatures along axis are plotted at right, temperature and velocity profiles along diameters are plotted at left

room temperature, is less than the corresponding percentage of change in the downward velocity.

The boundary of the jet is defined as the surface of zero downward velocity. The principal portion of the boundary shown in Fig. 8 was established by observing the travel of thin jets of smoke directed horizontally toward the boundary by a smoke gun. The boundary of the bottom portion of the jet could not be established satisfactorily by this technique. The shape of the bottom of the jet shown in Fig. 8 is, therefore, approximately the shape revealed by pictures of jets made visible by smoke. This shape is also similar to that previously developed from pitot-tube determinations of velocity. (Both the smoke gun and pitot-tube are uncertain means for determining the jet boundary[3].) As was previously pointed out, the bottom of the jet fluctuates about a mean position. The fluctuations occasionally take the form of surges.

The inverted cone that extends downward from the orifice in Fig. 8 is a region of the primary zone which has not been penetrated by the mixing process associated with entrainment. In this region the velocity of the jet deviates little

from its orifice value. Another cone might be conceived of as being similarly drawn, but with its apex somewhat closer to the orifice. Such a cone would represent the region in which the jet temperature likewise deviates little from its orifice value.

The jet should be regarded as being projected into an envelope of ambient air which has been set in motion and elevated in temperature by the jet itself. The jet entrains air from this envelope at a temperature above that of the remote room air, measured at a point approximately table height (30 in.) from the floor.

Limited data on the primary zones of short throw jets indicate that the downward penetration of this zone will be significantly reduced as the throw is reduced below a value of approximately 18 orifice diameters. These data also indicate that the temperature and velocity gradients in the vicinity of the orifice may be more pronounced in short throw than in long throw jets. Fig. 8 therefore, should be regarded as typifying long throw jets of the type under consideration, not jets of all types.

BASIC CONSIDERATIONS

By dimensional analysis, a dimensionless form of equation is obtainable for the downward throw of perpendicular jets of heated air. Let the assumption be made that the throw will be predictable from the following quantities:

D_o = diameter of the orifice;
$(\rho_o U_o^2)$ = momentum force of the jet at the orifice, per unit of orifice area;
$g\,\rho_o[(\rho_a/\rho_o) - 1]$ = buoyancy force per unit volume of orifice air.

Then, by dimensional analysis:

$$\frac{L_{max}}{D_o} = f_1\left(\frac{U_o^2/gD_o}{\frac{\rho_a}{\rho_o} - 1}\right)^*$$

Here, f_1 indicates some function, the form of which is to be determined. The quantity $[(\rho_a/\rho_o) - 1]$ may be replaced by $[(T_o/T_a) - 1]$ when the jet is a jet of air projected into a buoyant atmosphere with a temperature, T_a, measured at the point at which ρ_a is evaluated.

The dimensionless quantity

$$\left(\frac{U_o^2/gD_o}{\frac{\rho_a}{\rho_o} - 1}\right)$$

and quantities similarly formed, will be referred to as buoyancy numbers and

*The initial grouping of terms assumes that ρ_a and ρ_o will not form a separate dimensionless group. When this assumption is not made, dimensional analysis gives:

$$\frac{L_{max}}{D_o} = f\left\{\left[\frac{U_o^2/gD_o}{(\rho_a/\rho_o - 1)}\right], [\rho_a/\rho_o]\right\}$$

For the range of conditions studied and for the particular value of ρ_a employed, the quantity ρ_a/ρ_o was found not to be a significant factor. See discussion of ρ_a/ρ_o under curve fitting procedure in the Appendix.

designated by the symbol B with appropriate subscripts. Thus B_o will be evaluated at the orifice, and will have the value

$$\left(\frac{U_o^2/gD_o}{\frac{\rho_a}{\rho_o}-1}\right)$$

or its equivalent

$$\left(\frac{U_o^2/gD_o}{\frac{T_o}{T_a}-1}\right)$$

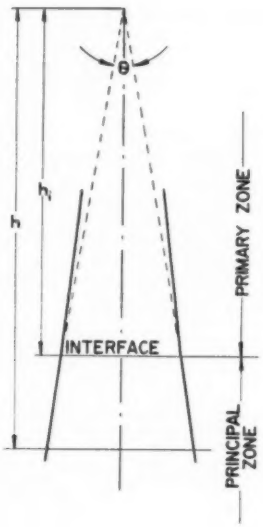


FIG. 9. TANGENTS TO BOUNDARY OF PRINCIPAL ZONE AT NOMINAL INTERFACE BETWEEN PRIMARY ZONE AND PRINCIPAL ZONE CONVERGE ON AXIS AND FORM ANGLE, θ

Using this symbolization, Fig. 3 indicates that:

1. For throws plotted in Fig. 3 equal to or greater than 8 orifice diameters, and for the range of conditions tabulated in Table 1,
$L_{max}/D_o = 1.66(B_o)^{1/4}$, $\pm$ 10%.

2. For throws less than 8 orifice diameters, and for the range of conditions tabulated in Table 1,
$L_{max}/D_o = 0.36\ B_o$, $\pm$ 10%.

The nominal interface between the primary zone and the principal zone may, like the orifice, be treated as a section of the jet at which a buoyancy number may be evaluated and used to predict the subsequent travel of the jet. When the interface is so treated, and certain assumptions are made in addition to

the kind previously employed, the following form of equation, developed in the Appendix, is obtainable:

$$\frac{\Delta L_{max}}{D_i} = \frac{C}{a}\left([ab_1 B_{ic}+1]^{\frac{1}{2}}-1\right) \quad \ldots\ldots\ldots\ldots \quad (3)$$

where

B_{1c} = the buoyancy number evaluated at the center of the interface between the primary and principal zones.

a = an apex angle and entrainment rate factor, and has the value 2 tan $\theta/2$, where θ is the apex angle of the principal zone as shown in Fig. 9.

For very low values of throw, one may assume that the principal zone begins at the orifice and that a approaches zero. Then, for this case:

$$\Delta L_{max} = L_{max}$$
$$D_i = D_o$$
$$B_{ic} = B_o$$

and,

$$\frac{L_{max}}{D_o} = (Cb_1/2)B_o = b_2 B_o \quad \ldots\ldots\ldots\ldots \quad (4)$$

For long-throw jets, a straight line relation between $\frac{D_1 B_{1c}}{D_o}$ and B_o will be assumed. When this is done, and substitutions are made in Equation 3 that are in accord with the geometry of Fig. 9, the following form of equation, which is developed as Equation 7A in the Appendix, is obtainable for use in matching experimental data:

$$\frac{L_{max}}{D_o} = \left[C_1 B_o + C_2\right]^{\frac{1}{2}} + C_3 \quad \ldots\ldots\ldots\ldots \quad (5)$$

Equations 1 and 2, given earlier in this paper, have the form:

$L_{max}/D_o = b_2 B_o$, for Equation 2, and
$L_{max}/D_o = [C_1 B_o]^{1/2}$, for Equation 1, (obtained by setting C_2 and C_3 each to zero).

The coefficients b_2 and C_1 were established by the method of least squares. Equations 1 and 2 plot as straight lines on log-log paper, with a slope of unity in the region of short throws and a slope of ½ in the region of long throws. For straight-line equations with these slopes on log-log paper, Equations 1 and 2 constitute the best match of experimental data.

The applicable range of values of C_1, C_2, and C_3 was explored in establishing the most advantageous modification of Equation 5.

COMPARISONS WITH DATA IN PREVIOUS PAPER

The present paper uses the subscript o, in place of subscript 1, to denote conditions at the orifice and introduces a new symbol, B, to denote the buoyancy

number. Except for these changes, the notation used in the present paper is the same as the corresponding notation used in the previous paper.[1]

Equation 1 for throw as given in the previous paper has the following form in the notation of the present paper:

$$\frac{L_{max}}{D_o} = \frac{1}{a}\left([aB_o+1]^{\frac{1}{4}}-1\right) \quad \ldots\ldots\ldots\ldots\ldots (6)$$

Equation 1 of the previous paper was derived on the basis of assumptions which were equivalent to placing the beginning of the principal zone at the orifice. When the principal zone is so placed, Equation 3 of the present paper leads directly to Equation 6 and therefore to Equation 1 of the previous paper.

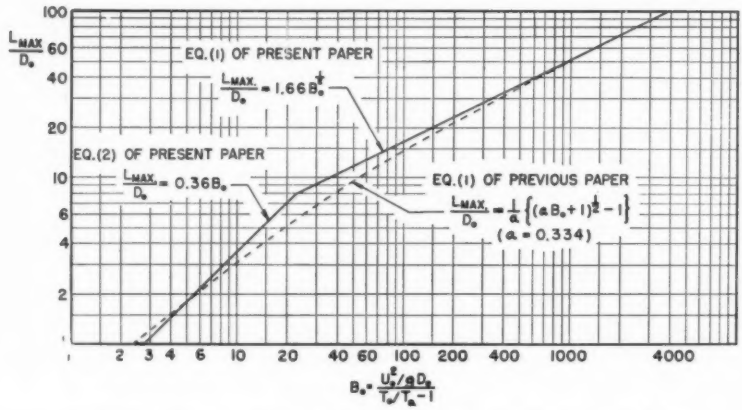


FIG. 10. COMPARISON OF PRESENT AND PREVIOUS EQUATIONS FOR EVALUATING THROW

Thus for this case L_{max} will replace ΔL_{max} in Equation 3, D_o will replace D_1, B_o will replace B_{1c}, and b_1 and C will each be unity. Fig. 10 shows a comparison of Equation 1 of the previous paper with Equations 1 and 2 of the present paper. For this comparison, the factor a has been given the value previously assigned to it, namely, 0.334.

Fig. 10 of the previous paper presents data on the upper and lower limits of the varying travel of a jet from a 5.75-in. converging nozzle. If these data were replotted on Fig. 3 of the present paper, they would form a curve of mean values that would lie parallel to but 10 percent below the mean curve established in Fig. 3 for throws in excess of 8 orifice diameters. Two considerations are involved here. First, the 5.75-in. convergent nozzle was not a long-radius *A.S.M.E.* nozzle, although it had somewhat the shape of that type of nozzle. Second, means for determining the bottom of the jet employed previous to 1948 were probably somewhat less sensitive than those employed since that date. However, aside from these considerations, the discrepancy between the mean values of the two sets of data is not significant. The mean of the upper and

lower limits of travel of a jet is not necessarily the same as the measured mean throw, and, in any event, the data plotted in Fig. 3 show a normal spread of ±10 percent.

CONCLUSIONS

Figs. 3 to 6 are the basic contributions of this paper to information on the downward throw of perpendicular jets of heated air. For the type of jets with which the paper deals, and for the range of conditions tabulated in Table 1, these figures show:

1. That B_o, the buoyancy number

$$\frac{U_o^2/gD_o}{(T_o - T_a)/T_a}$$

is an adequate parameter for correlating the data presented on the downward throw of perpendicular jets of heated air (For a discussion of T_o/T_a as an independent parameter, refer to Curve Fitting Procedure in Appendix);

2. That these data plotted against this buoyancy number on log-log paper form a band of most probable values;

3. That this band is satisfactorily represented by two equations:

(*a*) For throws between 8 and 90 orifice diameters,

$$L_{max}/D_o = 1.66\ B_o^{1/2},\ \pm\ 10 \text{ percent.}$$

(*b*) For throws less than 8 orifice diameters,

$$L_{max}/D_o = 0.36\ B_o,\ \pm\ 10 \text{ percent.}$$

The second of the foregoing equations (*b*) is supported only by a limited amount of data obtained on jets from a 4-in. nozzle. It should therefore be regarded simply as indicative of values of throw shorter than 8 orifice diameters. For throws between 8 and 45 orifice diameters, the first equation (*a*) is supported by a substantial amount of data on jets from nozzles having orifices 2 to 17 in. in diameter. For throws in excess of 45 orifice diameters, this equation is supported by data on jets from a 2-in. nozzle. Conservative use of the first equation (*a*) for throws in excess of 45 orifice diameters appears justified.

At least one additional study using a nozzle larger than 2 in. in diameter should be made to verify the data on throws in excess of 45 orifice diameters. Many more data are needed on throws below 8 orifice diameters. A standardized procedure for locating the bottom of a downwardly projected jet of heated air is needed, and should be developed. Until such a procedure is developed, results obtained in the field and in the laboratory may not be comparable. The procedure should, so far as practicable, make use of recording instruments to locate the bottom of the jet. The feasibility of making use of such instruments, therefore, is a matter that should be investigated promptly.

An investigation of the throw of jets cannot be divorced completely from a study of other aspects of jet behavior. The data presented in this paper therefore will be under continuous review as research on other aspects of the jet progresses. Except for this, and the additional data to be obtained for very short and very long throws, this paper fulfills one objective of the research program;

that of providing data for predicting the downward throw of heated jets from standard long-radius *A.S.M.E.* nozzles under the conditions investigated.

ACKNOWLEDGMENTS

The data presented in this paper are the result of observations made by many who previously were on the staff of the Department of Mechanical Engineering at Kansas State College as research assistants or instructors. Among these, Huo-Psi Pan, P. S. Wood, and E. H. Copeland made noteworthy contributions. The statistical analyses were made by Dr. H. C. Fryer of the Department of Mathematics, Kansas State College.

The guidance of the members of the Technical Advisory Committee on Air Distribution, and particularly the encouragement offered by Ernest Szekely, Howard Brinen and Dean M. A. Durland were exceedingly helpful.

REFERENCES

1. Downward Projection of Heated Air, by Linn Helander and C. V. Jakowatz (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 71).

2. The Effects of Boundary Conditions on the Primary Zone of a Downward Projected Jet of Heated Air, by G. E. Moeller (Master's Report, Kansas State College).

3. A Smoke-Filament Technique for Experimental Research in Room Air Distribution, by H. B. Nottage, J. G. Slaby, and W. P. Gojsza (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 399).

APPENDIX

DEVELOPMENT OF FORM OF EQUATION FOR THE DOWNWARD THROW

Referring to Fig. 9, let the point of convergence of the sides of the principal zone extended tangentially be regarded as the origin of a coordinate system. Assume that h_{max} is a function of:

1. A characteristic momentum force per unit area, $\rho_{ic}U_{ic}^2$;
2. A characteristic buoyancy force per unit volume,

$$g\rho_{ic}\left[\left(\frac{\rho_a}{\rho_{ic}}\right)-1\right];$$

3. A characteristic length, h_i, and
4. A dimensionless (velocity-profile)-(temperature-profile) shape factor, b_1.

Then dimensional analysis gives:

$$\frac{h_{max}}{h_i}=f_1\left\{b_1U_{ic}^2/gh_i\right\}\Big/\left\{\left(\frac{\rho_a}{\rho_{ic}}\right)-1\right\} \quad \text{. . . . . (1-a)}$$

Substituting $(\Delta L_{max}+h_1)$ for h_{max}, (D_1/a) for h_1, and solving for $(\Delta L_{max}/D_1)$:

$$\frac{\Delta L_{max}}{D_i}=\frac{1}{a}\left[f_1(ab_1\beta_{ic})-1\right] \quad \text{. . . . . (2-a)}$$

where

$$B_{ic} = [U_{ic}^2/gD_i] \Big/ \left[\frac{\rho_a}{\rho_{ic}} - 1\right].$$

Apply Equation 2*a* to the case of a body falling with negligible friction* and entrainment in a buoyant fluid and having an initial velocity U_{1c}. For this case a will approximate zero and ΔL_{max} will approximate

$$b_1 U_{ic}^2/2g \left[\frac{\rho_a}{\rho_{ic}} - 1\right]:$$

Substituting this value for ΔL_{max} in Equation 2*a* and transposing,

$$f_1[ab_1 B_{ic}] = \tfrac{1}{2} ab_1 B_{ic} + 1, \text{ for} \quad \ldots\ldots\ldots\ldots \quad (3\text{-a})$$

the case wherein a approaches zero.

Experimental evidence shows that Equation 3*a* is not valid when ($a\ b_1 B_{1c}$) is large relative to unity. We must seek, therefore, a form of equation that resolves into Equation 3*a* when ($a\ b_1 B_{1c}$) approximates zero, but is not identical with Equation 3*a*. One such equation is:

$$f_1[ab_1 B_{ic}] = [ab_1 B_{ic} + 1]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots \quad (4\text{-a})$$

Combining Equations 2*a* and 4*a*, and introducing an experimental factor, C;

$$\frac{\Delta L_{max}}{D_i} = \frac{C}{a}\left\{[ab_i B_{ic} + 1]^{\frac{1}{2}} - 1\right\} \quad \ldots\ldots\ldots\ldots \quad (5\text{-a})$$

Multiply both sides of Equation 5*a* by D_1/D_o, replace ΔL_{max} by $(L_{max} - L_1)$, and solve for (L_{max}/D_o). Then:

$$\frac{L_{max}}{D_o} = \frac{C}{a}\left\{\left[a\left(b_i \frac{D_i}{D_o}\right)\left(\frac{D_i}{D_o} B_{ic}\right) + \left(\frac{D_i}{D_o}\right)^2\right]^{\frac{1}{2}} - \left(\frac{D_i}{D_o}\right)\right\} + \frac{L_i}{D_o} \quad \ldots\ldots \quad (6\text{-a})$$

As an approximation, let $(D_1/D_o)B_{1c} = nB_o + m$, and consolidate quantities other than B_o into the form of factors to be experimentally determined. Then:

$$\frac{L_{max}}{D_o} = [C_1 B_o + C_2]^{\frac{1}{2}} + C_3 \quad \ldots\ldots\ldots\ldots \quad (7\text{-a})$$

CURVE FITTING PROCEDURE

Various modifications of Equation 7*a* were tried out for throws in excess of approximately 8 orifice diameters. None gave a significantly better match of experi-

---

*In the absence of friction the body will fall until the work done against the buoyancy force equals the initial kinetic energy of the body. Thus:

Travel × Buoyancy force = Initial kinetic energy

$(\Delta L_{max}) \times g \times [(\rho_a/\rho_{ic}) - 1] = b_1 U_{ic}^2/2$, where b_1 for this case is unity.

mental data than a straight-line equation on log-log paper. When no restrictions were placed on slopes, the method of least squares gave as the exponents of B_o:

For long throw jets—0.49 instead of 0.50.

For short throw jets—1.02 instead of 1.00.

To determine whether or not T_o/T_a, or ρ_a/ρ_o, should be included as a separate parameter in the equation for throw, the following procedure was employed:

Plotted values of L_{max}/D_o were organized to form groups within each of which the buoyancy number, B_o, had an approximately common value. The values of L_{max}/D_o in each group were then adjusted to a common value of B_o for the group, and these compensated values were plotted against $(T_o - T_a)/T_a$. The result, for each group, was a random scattering of points about a horizontal line. The quantity,

$$\frac{L_{max}}{D_o\sqrt{B_o}}$$

was also plotted against each of the quantities $[(T_o/T_a)\text{-}1]$, T_a, and T_o. The result in each case was again a random scattering of points about a horizontal line. These results indicate that for the range of conditions employed and for the location at which T_a was measured, neither T_o/T_a nor T_o is a significant independent parameter. Since the room temperature varies from floor to ceiling, T_o/T_a should be regarded as a probable independent parameter unless experimental evidence proves it to be otherwise. For air, T_o/T_a may be set equal to ρ_a/ρ_o. Therefore, when T_o/T_a is not a significant independent factor, ρ_a/ρ_o likewise will not be a significant independent factor.

1476

No. 1476

AIR VELOCITIES IN VENTILATING JETS

By G. L. TUVE*, CLEVELAND, OHIO

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the Case Institute of Technology.

THE ENGINEER needs answers to a few simple questions about a jet discharged from any air supply outlet. In general terms, the most common questions are:

1. How far does the jet travel or what is the throw?
2. What is the shape of its spread or coverage?
3. How can velocity at any selected point within the jet be determined?

The engineer admits that although he wants simple answers, the safest way to insure their accuracy is by developing as complete an understanding as possible of all jet behavior, based on theory and experiment. Both engineers and scientists have paid much attention to the free jet problem in the past 20 years. At least 50 papers have been published. Most papers of high theoretical excellence are unfortunately deficient in supporting experimental work to determine the constants of the equations. Conversely, experimentalists have not paid enough attention to theoretical analysis. After making a critical summary of more than 175 references, Nottage concluded, *the ventilation jet problem in air distribution appears to have attained a stage where careful experimentation and semi-empirical theories can be made highly productive of design data*[1].

It is this author's aim to select some of the promising semi-empirical theories, and support them by experimental evidence. Special attention is given to annular jets.

---

* Professor of Mechanical Engineering, Case Institute of Technology. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January 1953.

A *free jet* is one discharged into a relatively large room or other free atmospheric space. Limitations are imposed that no surfaces or objects are near enough to the stream to interfere with formation of the natural flow-pattern and that the primary stream is straight-flowing, free from pulsations and helical flow. Isothermal jets are first considered, but modifications for non-isothermal conditions usually encountered, are not too extensive.

The overall study includes two classes or groups of practical devices:

Class 1. Straight-flow devices; nozzle, orifices, slots, perforated panels, straight-flow grilles and registers and multiple openings located close together.

Class 2. Annular outlets without diffusers (including ceiling plaques).

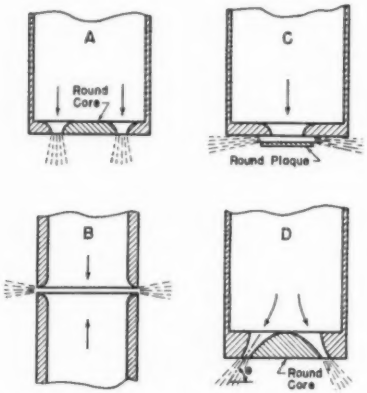


FIG. 1. CROSS-SECTIONS OF ROUND DUCTS WITH FOUR BASIC TYPES OF ANNULAR OUTLETS

TYPES OF ANNULAR JETS

Annular openings may be arranged to produce straight-flow axial jets or pure radial jets. Four basic types of annular outlets shown in Fig. 1 are: Outlet A, a circular slot in a wall or ceiling, discharging perpendicular to the same, and formed by a round nozzle with a solid core in the center; Outlet B, the other extreme, the jet is directed radially from the axis of the approach duct (This form amounts to a leakage ring between two identical ducts placed end to end.); Outlet C, usually called a ceiling plaque, a radial outlet with a plane boundary extending from one edge, might be the limiting condition of outlet B as the radial jet is brought close to a wall or ceiling, or equivalent to one-half of outlet B when the jet of the latter is bisected by a radial plane; Outlet D is the case between A and B, and may approach one or the other, depending on the Angle Θ.

There are in practice many variations of these fundamental forms, depending on the nature of the approach to the outlet, the shape and dimensions of the

outlet and plaque, and the presence of guiding vanes or surfaces. A variation of outlet D, not considered here, is the addition of an annular diffuser or expanding nozzle. Another whole family of devices is produced by using square or rectangular forms instead of circular.

FOUR ZONES IN A JET

When the several portions of a free air stream leave the outlet face in straight parallel flow, a surprising similarity has been found at a distance from the outlet, whether the latter is round, rectangular, grille-like or a perforated panel. Even with wide-angle grilles or annular outlets these similarities admit the use of the same type of appraisal and analysis.

Four major *zones* are recognized along the direction of flow of a free jet. These zones may be roughly defined in terms of the maximum or center-core velocity that exists at the cross-section being considered:

Zone 1: A short zone, of 2 to 6 *diameters* from the outlet face (Distance from the outlet discharge face is, for convenience, expressed in *diameters.* For all axial-flow outlets, this is the diameter of an equivalent round nozzle of the same total gross area.) Maximum or *core* velocity is very nearly equal to the original outlet velocity throughout this length.

Zone 2: A transition zone that usually extends to 8 or 10 *diameters.* Over a fair portion of this zone, the maximum velocity may vary inversely as the square root of the distance from the outlet. For a rectangular outlet of large aspect ratio this zone is elongated.

Zone 3: An extensive zone and one in which the maximum velocity varies inversely as the distance from the outlet. This zone is from 25 to 100 *diameters* long, depending on the shape and area of outlet and the initial velocity. This is often called the zone of fully established turbulence or the principal zone of the jet.

Zone 4: A terminal zone in which the residual velocity decays rapidly into a large-scale turbulence. Within a few diameters the maximum velocity subsides to the range below 50 fpm usually regarded as *still air.*

In application of air supply jets to ventilation and air conditioning the important part of the jet is Zone 3, in which center velocity varies inversely as the distance from the outlet. In Zones 1 and 2 of a jet from a rectangular outlet there is a transition to circular or elliptical cross-section. For axial-flow multiple or annular outlets the streams merge in Zones 1 and 2; in Zone 3 only the typical conical or elliptical stream is recognized. In Zones 1 and 2 measurement and accurate description of the stream may be difficult, due to rapid transitions, merging of jets, pressure variations and the small distances involved. In Zone 3, however, the cross-sectional velocity pattern or profile observed radially from any point along the center line of the jet resembles a typical *probability curve.*

ANALYTICAL APPROACH

A full analysis of the performance of a free jet not only involves the four *zones* already indicated, but should also be applicable to any one of the air-outlet combinations that are available to the engineer for air distribution in a space. In classifying air outlets the first consideration is the shape and size of openings. *Approach history* of the air stream must also be considered. The patterns of

velocity, direction, pressure and turbulence existing in the stream prior to or at the exit from the outlet face will have their effect downstream. The presence of guiding surfaces or vanes at or near the outlet, and the proximity of multiple jets must be taken into account.

Laws of continuity, conservation of momentum and conservation of energy, and also the Prandtl mixing length theory, have been applied to the analysis of free jets. Dimensional analysis and similarity have been useful tools. The vorticity transfer theory has been developed in a like manner to the momentum transfer theory. These have also been correlated with heat transfer and with mass transfer (diffusion by using heated or cooled air and introducing various diffusing gases). Some of these approaches have leaned heavily on the theories of turbulence. But turbulence is a most complex phenomenon and even when the concepts regarding it can be expressed mathematically, means of measuring the quantities experimentally are deficient. All the mathematical formulations defining jet performance call for experimental constants, and in many cases these are not as yet well established.

Analysis and experimentation are continuing, and the usefulness of each of the several theories is being explored by various investigators in this country and abroad. In the meantime, is it possible to combine the results of existing jet investigations into practical engineering methods for answering such questions as those stated in the first paragraph of this paper?

It appears that an engineering analysis, applied to the Zone 3 only, may be based on the assumption that the useful energy in a jet is represented by the velocities parallel to the jet axis. If these velocities can be determined with acceptable accuracy throughout the third zone the engineering problem may be solved. Other forms of energy in the jet are considered as *losses* and are taken into account only by empirical coefficients. For simplicity, the air is treated as an incompressible fluid and it is assumed that viscous flow is not encountered in Zone 3. These assumptions are usually within the limits of engineering accuracy.

Omission of Zones 1, 2 and 4 of the jet is justified both on the basis that data on these zones are of much less importance to the ventilating engineer, and that methods of measurement in these zones are as yet too involved and too tedious to be practical either in the engineering laboratory or in the field. A major difficulty in Zones 1 and 2 is the existence of static pressures either above or below atmospheric pressure.

DETERMINING THE CENTERLINE VELOCITY—AXIAL JETS

The two chief characteristics of Zone 3 of *established turbulent flow* are (1) maximum axial or centerline velocity is inversely proportional to the axial distance, (2) cross-stream velocity profiles are similar. A third characteristic of importance in non-isothermal jets is the similarity of temperature profiles.

Three common types of outlets will be considered.

Type 1—Round Outlets: More experiments have been made with round outlets than with any other kind, and the analytical treatment of the round jet is also the most satisfactory. The simple momentum transfer theory, (conservation of momentum), indicates that the relationship between the centerline velocity V_c and the distance X from the point source should be: $V_c X$ = a constant.

This Zone 2 *hyperbola* has also been recognized by most experimentalists, and has been verified by the fact that as V_c decreases with the distance from the

outlet, experimental points form a straight line at 45 deg on logarithmic coordinates. Examination of data reported by many investigators in this country and abroad shows that when non-dimensional logarithmic scales are used, even if the X-distance is measured from the discharge face, rather than from an imaginary point source (reported at about $-2.8D$ by Nottage[1], located at about $-2.3D$ for round nozzles by Ribner[2], and reported at $-2.0D$ by Schmidt[3]), a 45 deg straight line fits the experimental points over a wide range with good engineering accuracy. (To confirm these statements, data on round outlets are summarized in Table A-1 in the Appendix.) The proportionality constant designated as K or K' is defined by the common equation (see also THE GUIDE 1952, p. 650):

$$\frac{V_c}{V_o} = \frac{K'\sqrt{\frac{\pi}{4}D^2 \times C_d \times R_{FA}}}{X} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

which may be written

$$\frac{V_c}{V_o\sqrt{C_d R_{FA}}} = K'\frac{\sqrt{A}}{X} \quad \ldots\ldots\ldots\ldots\ldots (1a)$$

or

$$\frac{V_c}{V_o\sqrt{C_d R_{FA}}} = K\frac{D}{X} \quad \ldots\ldots\ldots\ldots\ldots (1b)$$

where

V_o = outlet velocity (at vena contracta), feet per minute.
V_c = centerline velocity in the free jet, feet per minute.
X = distance from outlet face, feet.
A = measured gross (core) area, square feet.
D = diameter of a circle equal to the gross area, feet.
C_d = coefficient of discharge of outlet, (usually between 0.65 and 0.90).
R_{FA} = proportion of free area to gross area, decimal.
K and K' = proportionality constant. In several previous papers, instead of using D, the equivalent diameter, $(A)^{1/2}$ has been used. Resulting values of K' would then be larger by a factor of $(4/\pi)^{1/2}$. Since rectangular streams tend to assume a circular shape in the third zone, the equivalent diameter furnishes a more logical unit in which to measure the distance X for free openings, but $(A)^{1/2}$ is more convenient for multiple openings, (where $A \times R_{FA}$ is the free area in square feet).

The limitations of Equations 1*a* or 1*b* have been fairly well indicated in reports already published. (For examples, see Appendix Table A-1.) Tests have been reported on outlet diameters from 0.25 in. to 12 in. and the outlet velocities were usually within the range 1000 to 10,000 fpm. For these ranges, Equation 1 apparently holds from about $X = 8D$ out to the point where V_c is of the order of 100 to 200 fpm.

It is generally accepted that the momentum or axial impulse force at the beginning of Zone 3 is less than that computed for the outlet face. It has also been demonstrated that Equation 1 holds even after the measured total momentum of the jet has been further reduced. A partial explanation may be that the core of the jet is not as readily affected by external conditions that cause

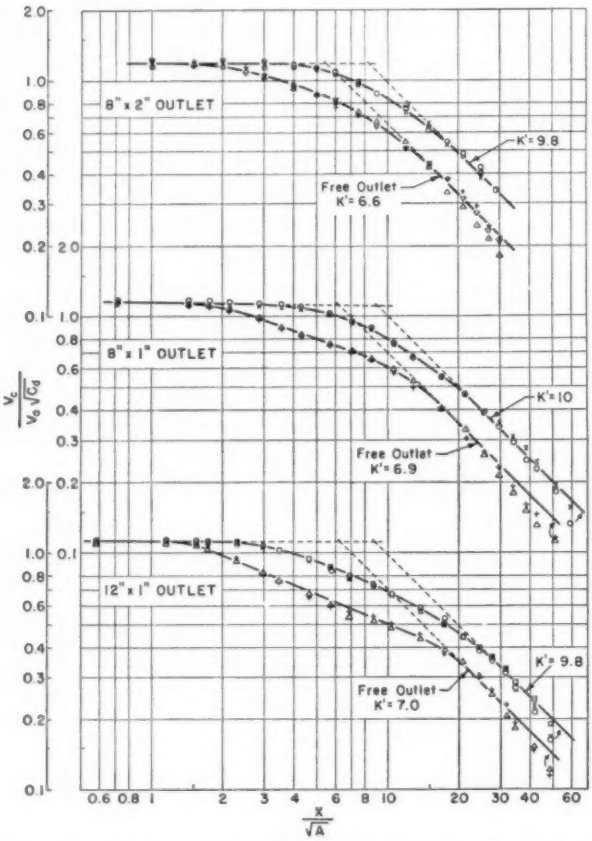


FIG. 2. EFFECT OF INCREASING ASPECT RATIO OF RECTANGULAR OUTLETS

Lower curve for free outlet; upper curve for outlet discharging along a wall or ceiling as in Fig. 10

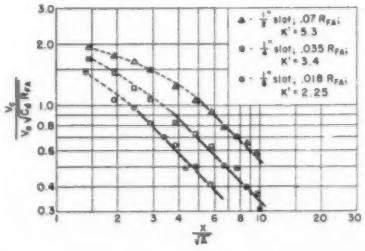


FIG. 3. CENTERLINE VELOCITIES IN THE JET FROM ANNULAR OUTLET, FIG. 1(A)

(O.D. = 14 in., V_o = 700 to 3300 fpm)

TABLE 1—RECOMMENDED VALUES OF THE CENTERLINE VELOCITY CONSTANT K OR K' (SEE EQUATION 1)

TYPE OF OUTLET	K		K'	
	V_o = 500 to 1000	V_o = 2000 to 10,000	V_o = 500 to 1000	V_o = 2000 to 10,000
Free Openings				
Round or Square	5.0	6.2	5.7	7.0
Rectangular, large aspect ratio (<40)	4.3	5.3	4.9	6.0
Annular slots axial or radial[a]	—	—	3.9	4.8
Grilles and Grids				
Free area 40 percent or more	4.1	5.0	4.7	5.7
Perforated Panels				
Free area 3 to 5 percent	2.7	3.3	3.0	3.7
Free area 10 to 20 percent	3.5	4.3	4.0	4.9

[a] For radial slots use X/H instead of $X/(A)^{1/2}$. H is the height or width of the slot.

Note: K and K' are indexes of loss in axial kinetic energy. Interpolate as required. Departures from maximum value indicate losses in first and second zones when compared with the jet from a rounded-entrance, circular nozzle.

loss of momentum near the perimeter. This zone of decreasing momentum involves a reduction in the rate of divergence of the jet.

While the dimensionless character of Equation 1 would indicate that the behavior of the jet is the same irrespective of the Reynolds number, several of the tests indicate that this is not strictly true. Slightly lower values of K are obtained for the lower outlet velocities and for very small diameters, indicated in Tables 1 and A-1. (For data and discussions on the Reynolds number effect see References 4 and 5).

A few studies of non-isothermal horizontal jets and jets of gases other than air, discharging to atmosphere, have indicated that the relationship of Equation 1 is not greatly affected by the difference in density, although the centerline of the jet will curve upward or downward.

Type 2—Rectangular Outlets: After full turbulent flow is established, *i.e.*, in Zone 3, the free jet from a rectangular outlet of low aspect ratio can hardly be distinguished from the jet from a round nozzle. Zone 2 is elongated as the aspect ratio is increased. With aspect ratios as high as 15 or 20, Zone 2 may extend to 20 or 30 diameters but there is a definite Zone 3 of greater length than Zone 2.

Theoretically, for an infinite slot the centerline velocity varies inversely as the square root of the distance, $[V_c(X)^{1/2} = C]$. Becher's test[6] on a 0.53 in. x 10.8 in. slot showed a square root relationship (Zone 2) between $X/(A)^{1/2} = 2$ and $X/(A)^{1/2} = 30$, with Zone 3 extending therefrom to about $X/(A)^{1/2} = 130$. Albertson[7], testing 6-ft long slots of widths 0.03 to 0.25 in. with guarded ends, (*i.e.*, discharging between parallel end walls), found the square root relationship was continuous to $V_o/20$, which amounted to a distance of some 2000 slot widths.

Results of experiments at Case Institute of Technology by W. Kerka show the increase in the length of Zone 2 as the aspect ratio is increased, Fig. 2. The outlets used in these tests were rectangular, rounded-entrance nozzles mounted

on a 3-ft cubical plenum chamber supplied from a metering duct. In these tests, the value of the throw constant K' is only slightly less than that for a round outlet, even when the aspect ratio is increased to 12. (See Appendix, Table A-2 for additional data.)

Type 3—Grilles, Grids, and Perforated Panels: The performance of multiple outlets can be correlated in terms of free area, and Equation 1 is still applicable for determining centerline velocities. Since the throw constant K is an index of the directional axial kinetic energy in the free jet in the third zone, it is expected that the values of K will be lower for multiple jets than for a single jet, especially when the jets are widely spaced as compared with their diameter. Grids and perforated panels of low free area (or high *grid density*) are such

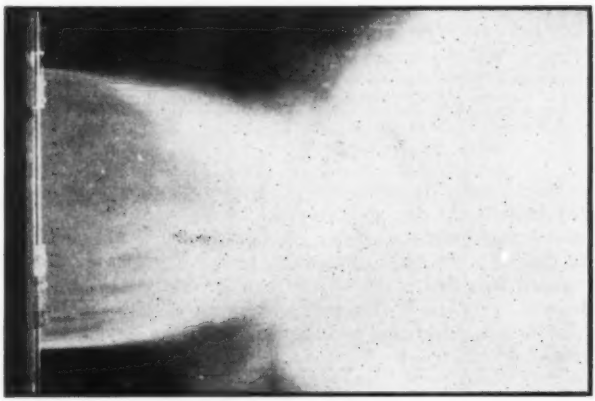


FIG. 4. SMOKE-PATTERN PHOTOGRAPH OF THE JET FROM ANNULAR OUTLET, FIG. 1(A)

($V_o = 3000$, slot 0.5 in. wide, 14 in. O.D.)

examples. Grid pattern and spacing should be related to the distance required for the formation of the coalesced, uniform jet, to the energy losses, and to the resulting value of K. When the free area is 50 or 60 percent or above, behavior is much like that of a free-open outlet of the same free area. Rydberg and Norback[8] state that for concentrated discharge openings of circular or rectangular holes and grids of various constructions, *the form of the holes or the distance between them does not matter*. Nottage[9] used two square-mesh grids of 44 percent free area in series as a turbulence producer and added to the turbulence by radial inward jets between the grids. He reported no difference in the Zone 3 centerline velocity with this combination, as compared with a free-open nozzle, except at very low outlet velocities. Values of K' from 4.8 to 6 are reported by Koestel[10] for three straight-flow bar grilles of 72 to 84 percent free area and a value of 6.2 for a perforated panel of 40 percent free area. Tests at Case showed a definite reduction from $K' = 6.2$ to $K' = 4.5$ when the outlet of an 8-in. nozzle was covered by a perforated panel of 9.1

percent free area, (see Figs. 3 and 6, Ref. 10). F. Tenelius reports[11] $K' = 4.5$ for a panel with 3.5 percent free area. These and other data indicate a throw constant K' in the range of 3.0 to 4.5 for multiple outlets of 3 to 10 percent free area, and 4.5 to 6 for greater free areas. The great variety of designs limits the accuracy of predictions.

Type 4—Annular Outlets, Axial Flow: An investigation of outlets of the type shown in Fig. 1(A) was made at Case Institute of Technology by C. K. Barton. He showed that with an annular slot of 14 in. O.D. such a ring-type jet converges and forms, within two or three diameters, a normal free jet (as in Zone 3) for which Equation 1 again applies. He correlated the values of K' on a free-area basis, and the results of tests on three outlets are shown in Fig. 3. Diminishing values of K' as the slot width and the ratio of free area

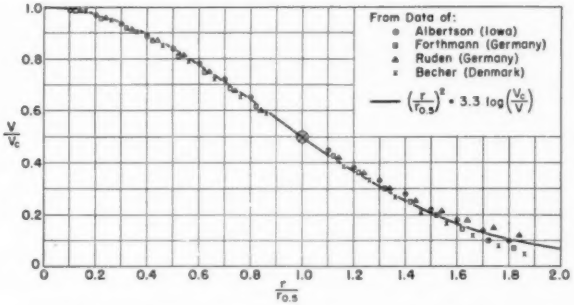


FIG. 5. CROSS-SECTIONAL VELOCITY PROFILES FOR STRAIGHT-FLOW TURBULENT JETS

to gross area is reduced, (same outside diameter), again indicate the large losses of directional kinetic energy in the formation of the coalesced jet.

Fig. 4 is a smoke-pattern photograph of the jet, showing successive convergence and divergence. Complete stream traverses were made for 14 combinations of slot area and outlet velocity. The average ratio of maximum centerline velocity to average volume velocity from these data was 2.5. The ring-type outlets, used by Barton, were rounded-entrance nozzles.

ANGLE OF DIVERGENCE OF AXIAL JETS

After computing the velocity at any desired point along the jet axis by Equation 1, it is required to find the cross-sectional jet pattern, *i.e.*, the velocity gradient or profile. One key to the cross-sectional dimensions is the natural discharge angle of a free jet. But here three difficulties are encountered: (1) if the outlet is not round or square, or if it is located near a wall or ceiling, the jet is not symmetrical; (2) the diameter of the jet decreases rapidly near the end of its throw; (3) the periphery of the jet is difficult to define and to measure because of the billowing turbulence encountered.

However, the natural free-jet angle is more persistent than might be expected. Either by smoke pictures, velocity measurements or theoretical computa-

tion of velocity profiles, several investigators have determined the angle of a circular jet. The early work of Zimm[12] for an air jet from a 2-in. open pipe, indicated a variation of jet angle with velocity, and a 20 deg angle at 1000 fpm. Using a hot-wire instrument and silk threads, he measured a 17 deg jet angle at 600 fpm and a 27 deg jet angle at 2400 fpm. Becher[6] measured smoke patterns from a variety of round nozzles and orifices 2 to 8 in. diameter, in the range 600 to 10,000 fpm and reported an average angle of about 24 deg, with but little variation due to type or size of outlet or face velocity. Ruden[13] reported a 16 deg angle for a heated jet from a 3-in. round nozzle. Albertson[7] showed test points to include a jet angle of about 20 deg, but the extreme edge of the jet formed a slightly greater angle. Nottage[14] measured jet angle by the

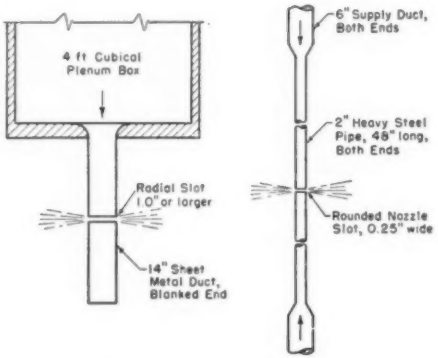


FIG. 6. TWO EXPERIMENTAL UNITS USED FOR STUDY OF RADIAL JETS

direction of the individual velocity vectors, using smoke indication, and found an average angle close to 20 deg, again irrespective of velocity over the range 100 to 6000 fpm outlet velocity. Jet-angle determinations previously reported from work at Case Institute of Technology have been based on velocity measurements with thermal and bridled-vane anemometers, and most of the results for round or square outlets have shown a jet angle close to 22 deg. The angle is very definite near the outlet, but it is easily affected by external drafts or by proximity of walls and other surfaces. Nottage, using a 6-in. nozzle, 7 ft from the ceiling in a 47 ft wide room, found that in elevation the 20 deg jet angle persisted only about 30 diameters while the plan view showed the 20 deg jet angle for a distance of almost 100 diameters.

Jet angles for other than square or circular outlets are about the same as the angle of a circular jet once the turbulent stream is fully established. Tests at Case[4] on 5 free-open slots 24 in. long and ¼ in. to 3 in. wide all gave measured jet angles of 22 to 23 deg. Other tests at Case on rectangular outlets of aspect ratio 5 or less, showed jet angles of 20 to 24 deg.

When multiple jets and a low percentage of free area are involved, the coalesced free turbulent jet usually diverges at an angle less than 20 deg.

Barton's data on axial jets from annular openings, Fig. 1(A) indicated a stream angle of about 18 deg in Zone 3.

JET VELOCITY PROFILES

The theoretical analyses of jet performance are based on the similarity of the curves of axial-velocity *vs.* radial-distance (perpendicular to the jet axis), so the cross-sectional velocity patterns should be both symmetrical and similar when measured from any point on the jet axis and they form a single curve on dimensionless coordinates. Within limits of experimental error, this has been confirmed in velocity traverses of free jets, (in Zone 3), by many experimenters.

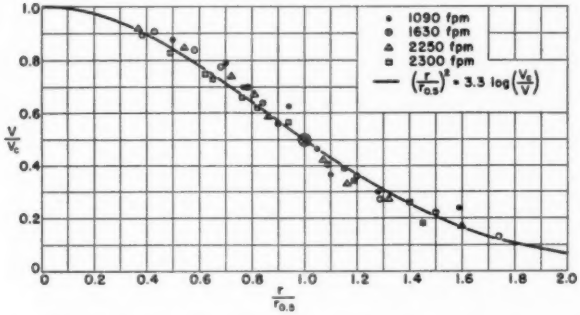


FIG. 7. CROSS-SECTIONAL VELOCITY PROFILE FOR A RADIAL JET FROM A 1.0-IN. SLOT IN A 14-IN. DUCT

(6 in. to 48 in. from outlet)

Cross-sectional velocity and temperature profiles for various heated, cooled and isothermal jets have been computed according to both the momentum-transfer theory (Prandtl-Tollmien), and the vorticity-transfer theory (Taylor-Goldstein). Modifications of both theories have been made to bring the results more closely into line with experimental traverses. Both Ruden[13] and Forthmann[15] made comparisons of their experimental velocity profiles with those computed by Tollmien's[16] momentum transfer theory. Howarth[17] concluded that the momentum-transfer theory predicted velocity profiles closer to the actual than did the others. Becher[6], Schmidt[3], Corrsin[18] and Nottage[5] also made comparisons of their experimental traverses with the theoretical predictions. General results of such studies seem to be that the modified theories predict the velocity profiles with fair accuracy, and that the velocity, temperature and density profiles are similar but do not coincide. Corrsin[18], Hinze[19] and others have shown experimentally that for horizontal jets the effects of temperature or density difference upon cross-jet velocity profiles are small.

It has been demonstrated by Albertson[7], Taylor[20], Keagy[21], Pai[22] and others, that the Gauss error-function curve (probability curve) is superior to the curves predicted from theory for practical matching of experimental results. Following this suggestion, Fig. 5 has been prepared from the composite experimental

results of Albertson[7], Becher[6], Ruden[13] and Forthmann[15], and these have been found comparable with data taken at Case Institute of Technology. This curve, of course, represents only one-half the jet, the other half being a symmetrical duplicate.

A convenient device, used in Fig. 5, for comparing both theoretical and empirical curves has been to determine at each cross-section of the jet the radius $r_{0.5}$ at which the axial velocity is one-half of the velocity at the centerline. All curves then pass through the same point at $V/V_c = 0.50$. The equation of the

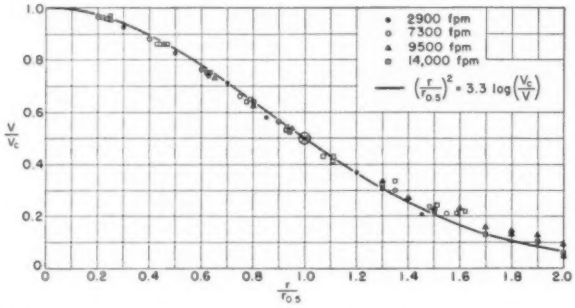


FIG. 8. CROSS-SECTIONAL VELOCITY PROFILE FOR A RADIAL JET FROM A 0.25 IN. SLOT IN A 2-IN. PIPE

(3 in. to 24 in. from outlet)

velocity profile, Fig. 5, is conveniently approximated by a simple equation using common logarithms:

$$\left(\frac{r}{r_{0.5}}\right)^2 = 3.3 \log \frac{V_c}{V} \quad \ldots\ldots\ldots\ldots\ldots (2)$$

where

r = the radial distance of the point under consideration, from the centerline of the jet.
$r_{0.5}$ = the radial distance in the same cross-sectional plane to the point where the velocity is one-half the centerline velocity.
V_c = the centerline velocity in the same cross-sectional plane.
V = the actual velocity at the point being considered.

Albertson's[7] data for the locations of $r_{0.5}$ indicated that the conical angle for this part of the jet was about 10 deg. Another investigator reported about 11 deg. It is apparent from Figs. 5, 7 and 8 that the conical angle for 0.5 V_c and $r_{0.5}$ is approximately one-half the total angle of divergence of the jet. It should be noted that experimental data may be used to express both $r_{0.5}$ and V_c in terms of X, the axial distance from the discharge face. For instance if the angle of the $r_{0.5}$ cone is 10.5 deg, $r_{0.5} = X \tan 10.5 = 0.185X$.

More work is needed to determine the true nature of the flow near the periphery of the jet. When the curve of Fig. 5 is used to determine velocities below 150 fpm, the result will probably be higher than the actual velocity in the stream

due to the effects of viscosity and the probability of external air currents. Therefore, at great distances from the outlet the error-function curve predicts higher velocities and an apparent wider spread of the jet than actually occurs.

While Fig. 5 is based on symmetrical jets with a constant angle of divergence, it may be used for engineering estimates for other jets. Most departures from the ideal symmetrical jet tend to produce lower velocities than those computed (from Fig. 5 or Equation 2) for the conical jet. Hence the computed velocity will be the maximum obtainable. A knowledge of this extreme condition is of great importance to the engineer.

For air streams from multiple outlets such as grilles and perforated panels, Koestel[10], Corrsin[23], and others have demonstrated that the velocity profiles of

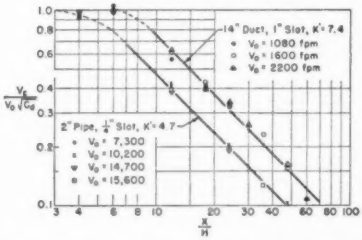


FIG. 9. CENTERLINE VELOCITIES IN JETS FROM RADIAL OUTLETS, FIG. 1(B)

the final jets in the third zone are very similar to those for a round nozzle, but the angles of divergence may be slightly different, usually smaller.

PERFORMANCE OF RADIAL JETS

The diameter of a conical jet increases directly as the axial distance from the outlet, (or, strictly speaking, from the apex) hence the area of the jet increases as the square of the axial distance. In a true two dimensional plane jet the area increases only as the first power of the axial distance. The radial jet, Fig. 1(B) might be called two-dimensional, but while its width (divergence) varies as the first power of the distance from the outlet, its circumference also varies directly as the distance from the outlet. Hence its area varies as the square of this distance, the same as that of a conical jet. It might therefore be expected that both the centerline velocity gradient and the cross-sectional velocity profiles in a radial jet would be similar to those of a round conical jet if the natural angles of divergence of the jets are nearly the same. Experiments have shown that such is the case.

Two sets of experiments on free radial jets have been conducted at Case Institute of Technology by K. Kjolner. In the first group, a 14-in. round duct was used, with air supplied from one end only, Fig. 6, and the outlet was a sharp-edged, continuous slot around the duct. In other words, the slot was an opening between two sheet metal ducts, the supply duct and a dead-end duct, each 20 in. long. In a series of tests from 1000 fpm to 2500 fpm outlet velocity the angle

of divergence of the jet was observed to be substantially constant at about 23 deg. These readings were made both with smoke (as described by Nottage) and by means of a sensitive null-type double impact tube device.

In the second test set-up, Fig. 6, the air was supplied from both directions, using 2-in. heavy pipe, and the discharge slot was accurately machined to standard flow-nozzle contour. In a series of tests with a slot width of 0.25 in. and various nozzle discharge velocities up to 15,000 fpm, the angle of the jet was again found to be practically independent of the initial jet velocity. Jet angles measured by smoke technique averaged 22 deg. From these two groups of tests, under widely differing conditions, it was concluded that free, radial, 360-deg jets assume the same natural angle of divergence as round, conical jets.

Complete 360-deg jet traverses were made of radial jets at various outlet velocities, using both units of Fig. 6. Thermal instruments (frequently cali-

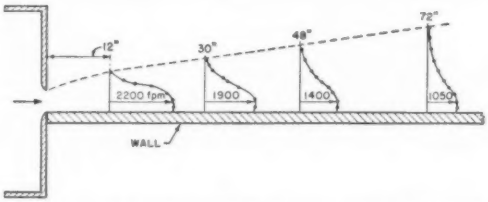


FIG. 10. JET VELOCITY PATTERNS FOR A SQUARE OUTLET DISCHARGING ALONG A WALL OR CEILING

($V_o = 2260$, $A = 0.10$, $C_d = 0.78$)

brated) were used for the lower velocities, and impact tubes for most of the measurements above 1000 fpm. Two of the important findings were that (1) the centerline velocity gradient in Zone 3 follows the hyperbolic law stated by Equation 1, and (2) the cross-jet velocity patterns or profiles at various distances from the outlet face are similar and are comparable with the profiles observed in a conical jet. Typical results in support of the latter conclusion are presented in Figs. 7 and 8.

Data on the centerline velocity gradient in these radial jets are given in Fig. 9. In this case the velocity ratio (on ordinates) has been plotted against X/H, where H is the height or width of the annular slot. Hence the constant K' is different from that defined by Equation 1, although numerical values are comparable. In effect this treats the jet as if it were being discharged from a square outlet of area H^2. The mathematical solution will be given later in a more complete report on the radial jet studies.

EFFECT OF WALLS AND CEILINGS

The jet explorations reported by Nottage demonstrated clearly that when the axis of a long jet is too close to the wall, floor or ceiling, and parallel with it, the spread of the jet in that direction is reduced. What about the extreme case where the wall or ceiling coincides with one edge of the outlet?

Five rectangular outlets were tested at Case by Kerka, both with and without adjacent walls. Data on the decrease of maximum jet velocity with distance

from the outlet have been given in Fig. 2 for three of the five outlets. The results clearly show a greater throw of the stream with the wall in place than was obtained from the same outlet discharging into free-open space. Additional data are given in Table 2. A graphic representation of the velocity patterns with the wall in place is given in Fig. 10. During all tests the maximum velocity remained very close to the wall surface. The angle of divergence of the jet perpendicular to wall was slightly less than one-half the angle of a free conical jet, while the angle of spread of the jet along the wall was greater than the divergence of a conical jet.

The longer throw of a jet along a surface is what should be expected from a careful analysis of the conditions. Data on sidewall friction in pipes and ducts and also tests of plates and airfoils immersed in a stream indicate that the friction drag between the jet and the wall is very small. Opinion is that

TABLE 2—TESTS OF RECTANGULAR ROUNDED-EDGE OUTLETS WITH AND WITHOUT ADJACENT WALL (KERKA)

OUTLET SIZE IN.	ASPECT RATIO	COEFFICIENT OF DISCHARGE	OUTLET VELOCITIES V_o, fpm	VALUES OF K', EQUATION 1_a	
				Free Outlet	With Wall, Fig. 10
4 × 4	1	0.782	1500 — 2600	7.0	8.9
8 × 2	4	0.819	1800 — 2750	6.6	9.8
8 × 1	8	0.883	2200 — 2800	6.9	10.0
12 × 1	12	0.897	1950 — 2750	7.0	9.8
16 × 1	16	0.883	1900 — 2600	5.4	7.3

the jet from an outlet of area A discharging parallel with a solid wall would be the equivalent of one half of a jet from an outlet with area 2A, bisected by a thin sheet through its axis. Since the maximum velocity at any cross section of a jet is proportional to the square root of the area, Equation 1, the value of K or K' for a jet adjacent to a wall should then be greater than that for a free jet by the factor $\sqrt{2}$. The jet should be wider as well as longer than a free jet. The conjectures have been confirmed by Becher[6] and also by the work of Kerka and Kjolner (see Table 2).

RADIAL JETS FROM CEILING PLAQUES

Studies of jets from ceiling plaques Fig. 1 (C) were made with 5.44-in. and 14-in. diameter supply nozzles, three plaque diameters, and slot widths from 0.5 to 4 in. Generally the jets were the same in all cases, with the maximum jet velocity persisting within less than 1 in. of the ceiling surface, even at considerable distances from the slot outlet, as shown in Fig. 11 and Table 3. This figure indicates that when a given volume of air is to be discharged from a plaque, the width of slot or distance of the plaque from the ceiling has little effect on the final jet pattern or velocities. This is strikingly illustrated by the more complete data of Table A-3 in the Appendix. This table shows further that the plaque diameter can be varied with only a small effect on the final stream, as regards either the maximum residual velocity at a distance from the

TABLE 3—RESULTS OF TESTS OF CEILING PLAQUES, 14-INCH NOZZLE, 14-INCH PLAQUE[a]

DISTANCE FROM OUTLET, INCHES	6	12	24	36	48
Slot Height, 1.0 in.—470 cfm					
V_c max, fpm	1400	1100	560	440	310
Distance, ceiling to V_c, in.	0.4	0.4	0.4	0.4	0.4
Width of jet, in.	2.0	2.7	3.7	4.5	5.3
Slot Height, 1.0 in.—500 cfm					
V_c max, fpm		1160	650		340
Distance, ceiling to V_c, in.		0.4	0.4		0.5
Width of jet, in.		2.7	3.4		4.0
Slot Height, 1.0 in.—703 cfm					
V_c max, fpm		1390	890	600	455
Distance, ceiling to V_c, in.		0.5	0.4	0.4	0.4
Width of jet, in.		3.2	4.6	5.3	6.1
Slot Height, 2.0 in.—1300 cfm					
V_c max, fpm		1050	950		
Distance, ceiling to V_c, in.		0.4	0.6		
Width of jet, in.		6.0	6.5		

[a] Data obtained by K. Kjolner.

slot or the spread of the jet. Moreover, the angle of jet divergence was less than 10 deg in all tests.

These data were summarized from full velocity traverses, involving in many cases the cross-sectional traverses and curve plotting on eight radii, 45 deg apart. With the equipment properly adjusted, there was little difference in the jet profiles in the various radial directions.

If this form of device is important enough to warrant full investigation, additional data will be required as a basis for accurate quantitative predictions.

APPLICATION TO ENGINEERING DESIGN

The foregoing summary of results of studies on axial and radial jets can be used as a basis for engineering design of supply outlets for a given space. The chief limitation would be that these data apply only to straight-flow axial or radial ventilating outlets, and not to outlets with guide vanes at the discharge face, or those that are expected to carry large heating or cooling loads.

Example 1—As a simple example in finding jet velocities, consider a 24 x 8 in. sidewall ventilating grille and assume that it has a gross (core) area of 1.0 sq ft, of which 70 percent is free area. It is delivering 850 cfm with its coefficient of discharge C_d being 0.80. Find (*a*) the distance from the outlet face at which the maximum velocity will have been reduced to 150 fpm, and (*b*) the velocity with which the jet will strike a structural column that is located 15 ft (perpendicular) from the grille and 42 in. to the left of the jet axis.

The outlet velocity at vena contracta is computed from the given data to be 1515 fpm. Then from Table 1, the value of K' is estimated to be 5.2. Since in question (*a*) the value of V_c is to be 150 fpm, the ratio $V_c/V_o(C_d R_{FA})^{\frac{1}{2}}$ is computed as 0.132. Then from Fig. 12 (Equation 1*a*) $X/(A)^{\frac{1}{2}}$ is 39.3.

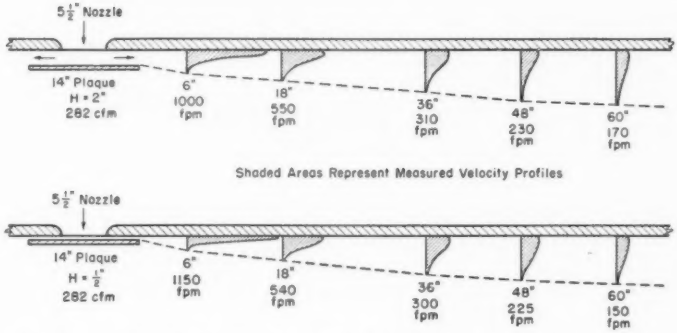


FIG. 11. AIR JETS FROM A 14-IN. CEILING PLAQUE FOR TWO SLOT WIDTHS WITH SAME CFM FLOW

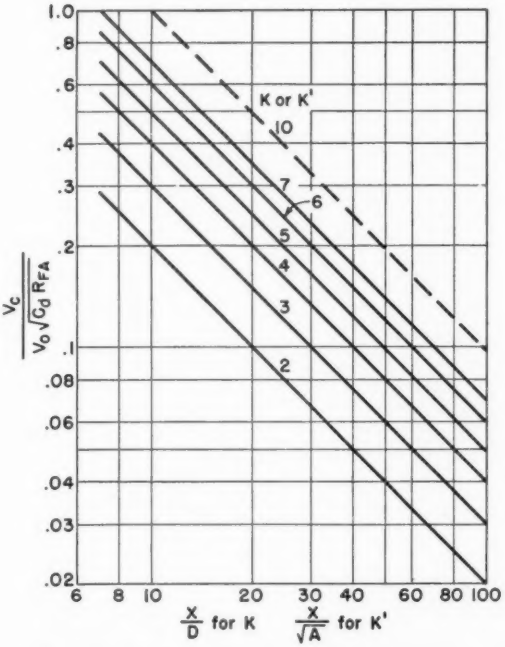


FIG. 12. ZONE 3 CENTERLINE VELOCITIES OF STRAIGHT AXIAL AND RADIAL JETS

Since $A = 1$, at approximately 39 ft from the grille the maximum velocity will not exceed 150 fpm.

For question (*b*) Equation 2 or Fig. 5 is used. It is required to find V, and r is given as 3.5 ft but $r_{0.5}$ and V_c must be computed. Again using Equation 1*a*, V_c is found to be 393 fpm. As previously stated, the jet angle of $r_{0.5}$ is about 10.5 deg, hence in this case $r_{0.5} = 15 \tan 10.5 = 2.78$ ft. Then $r/r_{0.5} = 1.26$, and from Fig. 5, $V/V_c = 0.33$. Hence the velocity of impingement at the column is 393 x 0.33 = 130 fpm.

CONCLUSIONS

Ventilating jets from annular outlets, either axial or radial may be analyzed by the same methods used for straight-flow, round jets. The hyperbolic law $V_cX = C$ is a close approximation for the centerline velocity gradient in the major zone of the jet in all cases except that of a very long slot, when the relation changes to $V_cX^{\frac{1}{2}} = C$. Cross-sectional velocity profiles are obtainable with engineering accuracy by a simple form of the probability curve, Equation 2, when the jet angles are assumed from reported data. A jet discharged along a wall or ceiling behaves like one-half a symmetrical jet and the maximum velocity occurs very close to the surface. Hence the throw and spread of such a jet are greater than those of a free jet.

As the procedures given herein for finding the jet velocities are based on several approximations, more accurate methods should be sought through further research. The two recommended equations must be used with caution for extreme axial and radial distances. Characteristics of the low-velocity regions are not yet well understood. Velocity patterns in closed rooms should be studied, with special attention to radial outlets, slots, perforated panels and other methods for circulating larger volumes of air without drafts. For both axial and radial jets, the effect of Reynolds number deserves more study, not only in Zone 3, but throughout the entire jet. The great variety of radial-jet outlets and the many operating conditions offer a large area for further investigation before the performance of all such jets can be predicted with confidence.

ACKNOWLEDGMENT

Acknowledgment is made for use of hitherto unpublished data from tests conducted at Case Institute of Technology by W. Kerka, former graduate student, C. K. Barton, former instructor in mechanical engineering, and K. Kjolner, former A.S.H.V.E. Research Fellow.

REFERENCES

1. Ventilation Jets in Room Air Distribution (1000 pages), by H. B. Nottage, Ph.D. Thesis, Case Institute of Technology, 1951. Loan copies of thesis available from A.S.H.V.E. Research Laboratory.

2. Field of Flow About a Jet and Effect of Jets on Stability of Jet-Propelled Airplanes, by H. S. Ribner (*N.A.C.A.* Wartime Report originally issued April 1946, as *Advance Confidential Report L6C13*).

3. Turbulente Ausbreitung eines Stromes erhitzter Luft, by W. Schmidt (*Zeitschrift fur angewandte Mathematik und Mechanik*, Vol. 21, 1941, pp. 265-278 and 351-363).

4. A.S.H.V.E. RESEARCH REPORT No. 1248—Control of Air Streams in Large Spaces, by G. L. Tuve and G. B. Priester (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, pp. 153-172).

5. A.S.H.V.E. RESEARCH REPORT No. 1443—Isothermal Ventilation-Jet Fundamentals, by H. B. Nottage, J. G. Slaby and W. P. Gojsza (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 107).

6. Calculation of Jets and Inlets in Ventilation (200 pages), by Poul Becher (Doctorate Thesis, Technical University of Denmark, Copenhagen, 1949. Danish with English summary).

7. Diffusion of Submerged Jets, by M. L. Albertson, Y. B. Dai, R. A. Jensen and Hunter Rouse (Iowa Institute of Hydraulic Research, June 30, 1947. Published in *A.S.C.E. Transactions*, 1948).

8. Air Distribution and Draft, by John Rydberg and Per Norbäck (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, pp. 225-240).

9. A.S.H.V.E. RESEARCH REPORT No. 1458—Outlet Turbulence Intensity as a Factor in Isothermal-Jet Flow, by H. B. Nottage, J. G. Slaby and W. P. Gojsza (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 343).

10. A.S.H.V.E. RESEARCH REPORT No. 1404—Comparative Study of Ventilating Jets from Various Types of Outlets, by Alfred Koestel, Philip Hermann and G. L. Tuve (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 459).

11. Private communication to author from F. Tenelius.

12. Uber die Stromungsvorgange im freien Luftstrahl, by W. Zimm (*Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, No. 234, Berlin, 1921).

13. Turbulente Ausbreitungsvorgange im Freistrahl, by P. Ruden (*Die Naturwissenschaften*, Vol. 21, 1933, pp. 375-378).

14. A.S.H.V.E. RESEARCH REPORT No. 1461—A Smoke Filament Technique for Experimental Research in Room Air Distribution, by H. B. Nottage, J. G. Slaby and W. P. Gojsza (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 399).

15. Turbulent Jet Expansion, by E. Forthmann (*Ingenieur-Archiv*, Vol. 5, 1934, pp. 42-54. Translation in *N.A.C.A. Technical Memorandum No. 789*, 1936).

16. Calculation of Turbulent Expansion Processes, by W. Tollmien (*Zeitschrift fur angewandte Mathematik und Mechanik*, Vol. 6, 1926, pp. 468-478. Translation in *N.A.C.A. Technical Memorandum No. 1085*, 1945).

17. Concerning the Velocity and Temperature Distributions in Plane and Axially Symmetrical Jets, by L. Howarth (*Proceedings Cambridge Philosophical Society*, Vol. 34, 1937, pp. 185-203).

18. Investigation of Flow in an Axially Symmetrical Heated Jet of Air, by S. Corrsin (*N.A.C.A. Wartime Report* originally issued December 1943 as *Advance Confidential Report 3L23*).

19. Heat and Mass Transfer in the Turbulent Mixing Zone of an Axially Symmetrical Jet, by J. O. Hinze and B. G. van de Hegge Zijnen (*Journal Applied Scientific Research*, Vol. 1, Sec. A, No. 5-6, 1949, pp. 435-461).

20. Turbulent Mixing in an Isothermal Jet, by J. F. Taylor, H. L. Grimmett and E. W. Comings (paper presented to *American Institute of Chemical Engineers*, 1949; see also *Chemical Engineering Progress*, No. 4, April 1951, pp. 175-180).

21. A Study of Freely Expanding Inhomogeneous Jets, by W. R. Keagy and A. E. Weller (*Proceedings Heat Transfer and Fluid Mechanics Institute*, University of California, 1949. Published by *American Society of Mechanical Engineers*, pp. 89-98, 1949).

APPENDIX

TABLE A-1—CENTERLINE VELOCITY DATA, ROUND OUTLETS

REPORTED BY	REFERENCE	OUTLET DIAMETER INCHES	OUTLET VELOCITY RANGE FPM	MEASURED RANGE IN 3RD ZONE (DIAMETERS)	VALUE OF K	VALUE OF K'
Cleve (Germany)	25	1.2, 2.0	4700 — 10600	20-100	6.0	6.8
Becher (Denmark)	6	2, 4, 8	900 — 4900	8-90	6.1	6.9
McElroy	24	5 (pipe)	7500	7-20	6.1	6.9
Albertson	7	0.25-1.00	11,200	7-250	6.2	7.0
Taylor	20	0.5	3000 —18000	8-30	6.2	7.0
Nottage	5	6	2500 — 6000	9-75	6.7	7.5
Nottage	5	6	1000	9-40	6.1	6.9
Nottage	5	6	500	8-20	5.1	5.8
Tuve	4	6, 8, 12	5000	25-100	6.2	7.0
Koestel	10	2.5, 8.2	700 — 1700	20-100	6.2	7.0
Koestel	10	8.7, 12 (orifices)	1300 — 7100	20-120	5.8	6.6
Koestel	10	2.5 (orifices)		50-125	5.1	5.8
Ruden (Germany)	13	2.8	(Heated Air)	6-20	6.1	6.9

Note: Outlets are round-edge nozzles unless otherwise specified.

22. Two Dimensional Jet Mixing of a Compressible Fluid, by S. I. Pai (*Journal Aeronautical Sciences, August*, 1949, pp. 463-469).

23. Investigation of the Behavior of Parallel Two-Dimensional Air Jets, by S. Corrsin (*N.A.C.A. Wartime Report* originally issued November 1944 as *Advance Confidential Report 4H24*).

24. Air Flow at Discharge of Fan Pipe Lines in Mines, by G. E. McElroy (*U. S. Bureau of Mines Report of Investigations No. 3730*, November, 1943).

25. Die Wirkungsweise von Wirbelluftdusen, by K. Cleve (*Feuerungstechnik*, Vol. 25, 1937, p. 317).

TABLE A-2—CENTERLINE VELOCITY DATA, FOR SQUARE AND RECTANGULAR OUTLETS

REPORTED BY	KOESTEL	BECHER	BECHER	BECHER	ALBERTSON	KERKA
Reference	10	6	6	6	7	Fig. 2
Outlet Sizes, in.	8 × 8 10 × 10	2 × 4	0.53 × 10.8	1.4 × 3.8	0.032 × 72 0.25 × 72	4 × 4 8 × 2 8 × 1
Aspect Ratio	1	2	20	2.7	Infinite[a]	1 to 8
Outlet velocity range, fpm	500 — 1600	3200 — 5100	5250	5150		
Zone ranges, $X/(A)^{1/2}$						
1st zone			0 — 1.5	0 — 5	0 — 5H	0 — 2
2nd zone			1.5 — 30	5 — 8	5 — 2000H	2 — 15
3rd zone	10 — 40	8 — 80	30 — 130	8 — 120	None	15 — 30
Value of K	4.8	7.2	5.3	6.0	[a]	6.0
Value of K'	5.4	8.1	6.0	6.8	[a]	6.8 (av)

[a] Long slot with guarded ends to produce two-dimensional jet.

TABLE A-3—RESULTS OF TESTS OF CEILING PLAQUES, 5-7/16-IN. NOZZLE THREE PLAQUE DIAMETERS, 6 SLOT WIDTHS, SAME CFM. DATA OBTAINED BY W. KERKA

DISTANCE FROM OUTLET, IN.	2	6	12	18	24	36	48	60	72
5-7/16 in. Nozzle, 5-7/16 in. Plaque, 282 cfm									
Slot Height, 0.5 in.									
V_c max, fpm	2300	1600	975	590	470	300	210	150	130
Distance, ceiling to V_c, in.	0.2	0.2	0.5	0.5	0.5	0.5	0.5	1.0	1.0
Width of jet, in.	1.3	1.5	2.5	3.0	3.5	4.5	5.0	6.0	6.5
Slot Height, 1.0 in.									
V_c max, fpm	1800	830	700	510	400	280	200	140	120
Distance, ceiling to V_c, in.	0.5	0.5	0.2	0.5	0.5	0.7	0.5	1.0	1.0
Width of jet, in.	2.7	3.5	3.0	3.5	3.5	5.5	5.5	6.0	6.0
Slot Height, 1.5 in.									
V_c max, fpm	1700	860	—	440	420	260	160	140	120
Distance, ceiling to V_c, in.	2.0	3.0	2.0	0.5	0.7	1.0	1.0	1.0	1.0
Width of jet, in.	4.0	6.0	7.0	5.0	4.5	5.0	6.0	6.0	6.0
5-7/16 in. Nozzle, 14 in. Plaque, 282 cfm									
Slot Height, 0.5 in.									
V_c max, fpm	1600	1150	700	540	410	300	225	150	130
Distance, ceiling to V_c, in.	0.2	0.2	0.5	0.5	0.5	0.5	1.0	1.0	1.0
Width of jet, in.	1.5	2.0	3.0	3.5	4.0	5.0	5.5	6.0	7.0
Slot Height, 1.0 in.									
V_c max, fpm	1200	875	630	480	400	280	210	150	120
Distance, ceiling to V_c, in.	0.5	0.4	0.5	0.5	0.5	0.5	1.0	1.0	0.5
Width of jet, in.	2.0	2.5	3.0	3.5	4.0	5.0	6.5	7.0	7.0
Slot Height, 2.0 in.									
V_c max, fpm	1000	1000	680	550	440	310	230	170	140
Distance, ceiling to V_c, in.	1.0	0.2	0.5	0.5	0.5	0.8	0.6	1.0	1.0
Width of jet, in.	2.5	3.0	3.0	3.0	3.5	5.0	5.5	6.0	6.5
5-7/16 in. Nozzle, 24 in. Plaque, 282 cfm									
Slot Height, 0.5 in.									
V_c max, fpm	1000	750	450		250		120		100
Distance, ceiling to V_c, in.	0.2	0.2	0.2		0.5		0.5		0.5
Width of jet, in.	1.5	1.7	3.0		5.0		6.0		6.0
Slot Height, 1.0 in.									
V_c max, fpm	700	500	350		300		170		110
Distance, ceiling to V_c, in.	0.3	0.4	0.2		0.4		0.6		1.0
Width of jet, in.	2.0	3.0	3.5		5.0		7.0		7.5
Slot Height, 1.5 in.									
V_c max, fpm	450	430	340		230		125		70
Distance, ceiling to V_c, in.	0.2	0.4	0.4		0.4		0.5		1.0
Width of jet, in.	2.2	2.5	3.0		4.1		5.5		6.5
Slot Height, 2.0 in.									
V_c max, fpm	425	460	360		240		120		85
Distance, ceiling to V_c, in.	0.4	0.4	0.4		0.5		0.4		0.5
Width of jet, in.	2.2	2.5	3.0		4.0		6.0		6.5
Slot Height, 3.0 in.									
V_c max, fpm	500	560	450		280		145		90
Distance, ceiling to V_c, in.	1.5	0.2	0.3		0.5		1.0		1.0
Width of jet, in.	3.2	3.0	3.5		4.0		5.0		5.0
Slot Height, 4.0 in.									
V_c max, fpm	600	550	475		270		150		90
Distance, ceiling to V_c, in.	2.5	0.1	0.5		0.5		0.8		0.5
Width of jet, in.	4.0	3.0	3.0		3.5		5.5		6.0

DISCUSSION

H. B. NOTTAGE, Los Angeles, Calif. (WRITTEN): Milestones are worth noting. This paper gives an application-minded condensation of extensive experimental work. By now we know how to treat isothermal, free-space jets, limiting interest to velocities above 50 fpm. Where data are not yet available, we know how to obtain them. Priority of time and effort is the major consideration.

What of the future? Emphasis is upon non-isothermal air distribution in confining spaces. The means are at hand to really grasp these problems. The keys are now available for a systematic, simplified approach; careful, persistent experimentation; judicious use of fundamental principles; and artful, engineering-wise empiricisms will provide the information.

To be specific, the profession needs:

1. Means of estimating maximum jet velocities. A few guiding experimental data will provide this.
2. Means of estimating jet boundary envelopes. The velocity boundary is best defined as the surface of zero mean-velocity component in direction of the maximum (or axial) jet velocity. Temperature boundary will differ from the velocity boundary.
3. To apply the principle of conservation of mass within the confining spaces. This will establish the balance of flow rates between various identifiable streams.
4. Magnitudes of the eddy viscosity and the turbulent Prandtl Number. These will establish cross-stream velocity and temperature profiles.
5. Application of the principle of energy conservation. This refers particularly to thermal energy with non-isothermal flow.
6. Application of momentum-principle force and moment balances. It should be remembered that the axial component of jet momentum flow rate is constant *only* when the discharge space is infinite.
7. Faith in abilities, willingness to be meticulous but practical in much-needed experimentation, and the far-sighted courage to finance this type of research.

CYRIL TASKER, Cleveland, Ohio: A.S.H.V.E. members have seldom listened to a more interesting presentation than that given by Professor Tuve. His questions are very important. When the Technical Advisory Committee on Air Distribution established a subcommittee to make an evaluation of air distribution research conducted by the Society during the past 20 years, it was working under a directive from the Committee on Research which said in effect, *Let's have a look at what we have been doing for 20 years! How valuable is it?*

The thousand page document mentioned by Professor Tuve, a masterly doctorate dissertation by H. B. Nottage, was evaluated along with other material. At first it was planned that all past research work be evaluated by a committee of college professors; later, the many advantages in having the work evaluated by a committee of practicing practical engineers became apparent. The evaluation subcommittee was composed of a group of men from the industry who have to use the data developed by research. This group presented very important recommendations to the Society. The first, a statement, was that the work Dr. Nottage did (much of it at the Laboratory), constitutes a real contribution to fundamental knowledge.

The committee recommended that sponsored air distribution work be continued at Kansas State College, Case Institute of Technology and Michigan State College.

Another recommendation was that a simple story be written about what has been done to acquaint members of the Society, otherwise interested, of the work's significance.

The committee also recommended thorough revision of chapters on Air Distribution and Air Duct Design in THE GUIDE 1952.

Field engineers should tell their application problems to the Research Committee and let its members know the type of information they want in THE GUIDE; also how they want technical papers written.

1477

No. 1477

COOLING A SMALL RESIDENCE WITH A TWO-HORSEPOWER MECHANICAL CONDENSING UNIT†

By H. T. GILKEY*, D. R. BAHNFLETH**, AND R. W. ROOSE***, URBANA, ILL.

PREVIOUS INVESTIGATIONS[1,2,3,4,5,6,7,8,9] in summer cooling were conducted in Warm-Air Heating Research Residence No. 1 in the summers of 1932 through 1938 and 1940. Since Research Residence No. 1 was a two and one-half story structure with a larger volume of storage effect than the smaller more typical home of post-war construction, it seemed advisable to continue the summer cooling studies in Research Residence No. 2.

The principal objectives of this investigation were:

1. Determination of the cooling load and its hourly variation when cooling Research Residence No. 2 as a whole, except the basement, under both night and day conditions.
2. Determination of the time lags in heat flow through the walls and ceiling.
3. Comparison of the actual cooling load with the cooling load calculated using the procedure outlined in THE GUIDE 1952.[10]
4. Determination of the operating characteristics of a two-ton mechanical refrigeration unit when outdoor air was introduced into the structure through the unit at the rate of one air change per hour.

RESEARCH RESIDENCE NO. 2

The residence was a one-story structure of frame construction with a large amount of glass exposure and with a full basement. The south exposure of the residence is shown in Fig. 1. The exposed wall section consisted of cedar

---

† This investigation was a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association.*

* Research Associate in Mechanical Engineering, University of Illinois. Junior Member of A.S.H.V.E.

** Research Assistant in Mechanical Engineering, University of Illinois.

*** Research Assistant Professor of Mechanical Engineering, University of Illinois. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the 59th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, January 1953.

shingles, 20-lb felt building paper, shiplap sheathing on 2 in. x 4 in. studdings, 3⅝-in. mineral wool blanket insulation with vapor barrier attached, and ¼-in. plywood panels on the interior. These panels were removable to permit access to the stud spaces for changing the duct system or for installation of additional instrumentation. The calculated coefficient of heat transmission, *U*, for this wall section was 0.07 Btu per hr (sq ft) (F). All windows and doors were weather-stripped and the windows on the east and west exposures were equipped with canvas awnings. Except for one large picture window in the living room, which was fixed in place and consisted of two panes, the windows were single glazed and of the horizontal sliding type. The doors were of conventional wood and glass construction. The south exposure of the residence was shaded by a 3-ft 10-in. roof overhang. The cooling load on the structure, except for the windows, was calculated by the sol-air temperature method given in THE GUIDE 1952[10], and the recommended method using the tables given in THE GUIDE for

FIG. 1. SOUTH VIEW OF RESEARCH RESIDENCE NO. 2

calculating the heat gain through the windows was used. As recommended in THE GUIDE 1952, the infiltration load was based on a wind velocity of 10 mph and the actual lineal feet of crack around the doors and the windows.

The residence included several special features of construction to make possible future investigations in warm-air heating. Open-web steel joists were used in the floor and ceiling so that warm air could be circulated in either joist space to permit investigation of the performance of a heating system using floor or ceiling panels. For this summer cooling investigation, however, the ceiling joist space was filled with 5 in. of mineral wool insulation. The calculated coefficient of heat transmission, *U*, for this ceiling was 0.07 Btu per hr (sq ft) (F).

The conditioned space consisted of all first-story rooms. Table 1 gives a summary of the room dimensions and volumes, and a compilation of the calculated cooling load for each room is given in Table 2. The cooling load calculations were based on outdoor design conditions of 95 F dry bulb, 76 F wet bulb, and indoor conditions of 75 F dry bulb, 62.5 F wet bulb (50 percent relative humidity). The maximum calculated cooling load on a design day for the structure with one air change per hour (120 cfm) of ventilation air mechanically introduced was 25,021 Btu per hr at 3:00 p.m. CST. The maximum calculated

TABLE 1—DATA FOR RESEARCH RESIDENCE NO. 2

A. Heat Transmission Coefficients, Btu per hr (sq ft) (F)	*U*
Insulated Frame Wall, with 3⅝-in. mineral wool insulation	0.07
Insulated Ceiling, with 5 in. mineral wool insulation	0.07
Outside Door—Front Hall	0.51
B. Infiltration Factors, cu ft per hr (ft of crack)	*I*
Door, weatherstripped	35
Window, weatherstripped	13
Fixed Window in Living Room	8

ROOM	DIMENSIONS				CEILING AREA[a]	EXPOSURE	NET WALL AREA	GLASS AREA	VOLUME
	Ft	In.	Ft	In.					
Living Room	21	10 ×	13	4	308	East	117	—	2,480
						South	89	100[b]	
South Bedroom	13	4 ×	11	0	169	South	71	28	1,325
&	4	0 ×	2	3		West	93	25	
South Bedroom Closets (2)	4	6 ×	1	11	23	South	19	—	147
Bath	7	8 ×	4	11	44	West	36	11	320
North Bedroom	11	11 ×	10	4	135	West	66	25	1,050
						North	61	30	
North Bedroom Closet	5	10 ×	2	4	18	North	24	—	116
						East	37	—	
Hall to Bath	6	7 ×	5	2	43	—	—	—	289
Front Hall	11	6 ×	4	7	59	North	22	21 (door)	448
Front Hall Closet	4	0 ×	2	4	9	—	—	—	79
Kitchen—Dinette	19	0 ×	11	4	232	North	124	42	1,830
						East	89	12	
	Total First Story				1,040	—	848	273	8,084

[a] Ceiling area includes area of partition and exterior walls.
[b] South Living Room glass area includes area of outside door having a high glass to wood area ratio.
Note: Ceiling Height of First Story—8 ft 6 in.

cooling load on a design day for the structure with infiltration air only was 20,808 Btu per hr at 3:00 p.m. CST.

The residence was completely furnished but unoccupied during this investigation.

EQUIPMENT AND INSTRUMENTATION

Cooling Unit: The cooling unit was one of two sections of a year 'round air conditioner which was installed in the basement of the residence. The conditioner occupied a floor area of 28 in. x 56 in. The furnace had a welded steel heat exchanger, and a burner sized for a rated input of 100,000 Btu per hr. The forward-curved, multiblade centrifugal fan, hereafter referred to as a blower, was equipped with a 10-in. wheel and was driven by a one-half horsepower motor and was integrally mounted in the cooling unit. The blower was rated to deliver 900 cfm, at standard air conditions, against a static pressure of 0.40 in. of water

TABLE 2—COMPILATION OF CALCULATED COOLING LOAD FOR RESEARCH RESIDENCE NO. 2

A. Maximum Load (Conducted and Transmitted) at 3:00 p. m. CST, Btu per hr

ROOM	WINDOWS	EXTERIOR WALL	CEILING	TOTAL
Living Room	3,365	201	940	4,506
South Bedroom	3,035	230	586	3,851
South Bedroom Closets (2)	[a]	[a]	[a]	[a]
Bath	880	36	256	1,172
North Bedroom	3,110	156	505	3,771
North Bedroom Closet	[a]	[a]	[a]	[a]
Hall to Bath	[a]	[a]	[a]	[a]
Front Hall	214 (door)	18	180	412
Front Hall Closet	[a]	[a]	[a]	[a]
Kitchen-Dinette	1,447	268	709	2,424
Totals	12,051	909	3,176	16,136

B. With Ventilation Air

	SENSIBLE	LATENT	TOTALS
Heat Gain to Rooms	16,136	—	16,136
Load of Blower	700[b]	—	700
Ventilation Load	2,590	3,320	5,910
Total for System			22,746
10 percent Miscellaneous Gains			2,275
	Grand Total		25,021

C. With Infiltration but with No Ventilation Air

	SENSIBLE	LATENT	TOTALS
Heat Gain to Rooms	16,136	—	16,136
Load of Blower	700[b]	—	700
Infiltration Load	913	1,167	2,080
Total for System			18,916
10 percent Miscellaneous Gains			1,892
	Grand Total		20,808

[a] Cooling load for these rooms included with larger adjoining rooms.
[b] Measured load.

external to the unit. The return-air was drawn into the unit and through the filter and the evaporator coil to the blower. The air was then forced through the furnace and into the duct system. The elements within the cooling unit were arranged in such a manner that the filter and the evaporator coil were located at the return-air entrance into the conditioner. This coil and the blower were in an insulated compartment and were therefore isolated from the compressor and condenser, which were located in the lower part of the unit.

The refrigeration capacity of the unit was 23,800 Btu per hour with *A.S.R.E.* standard inlet air conditions of 80 F dry bulb and 67 F wet bulb at rated air delivery of 900 cfm and at 120 psi refrigerant head pressure. The semi-hermetic compressor was directly connected to the 220 volt single-phase 2 hp motor. Both the motor and the shell-and-tube type condenser were water cooled. The refrigerant used was dichlorodifluoromethane.

Duct System: The duct system used in this cooling investigation was the same one used previously in heating investigations[11]; it was of the extended-plenum type, having uniformly sized trunk ducts leading from the furnace bonnet toward

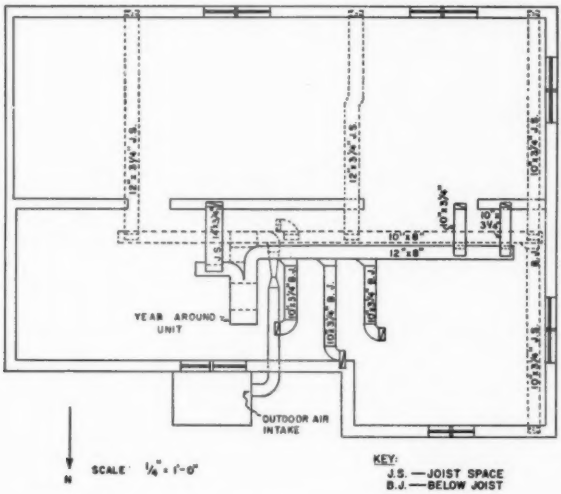


FIG. 2. BASEMENT PLAN SHOWING DUCT INSTALLATION

the east and west ends of the basement. Fig. 2 shows the basement plan and the duct installation. The branch ducts were connected to the top or side of the trunks and were unchanged in size from the trunk take-off fitting to the register stack-head. All registers in the first-story rooms were at the high-sidewall location, 6½ ft from the floor, with the exception of the baseboard register in the front hall near the door. Fig. 3 shows the first-story plan and the register locations. All return-air intakes were located in the baseboard. The system was designed [12] in accordance with Manual 7, of the *National Warm Air Heating and Air Conditioning Association.*

Instrumentation: To measure temperatures, approximately 180 thermocouples of 24-gage copper-constantan wire were installed. Thermocouples were placed at four different levels (floor, sitting, breathing, and ceiling) on standards located near the centers of each of the first-story rooms and at three stations in the basement. The location of the temperature measuring stations in the first-story rooms are shown in Fig. 3. Thermocouples were also installed in the

ceiling and floor surfaces, on the surface and within the exterior walls, in the attic, in the duct system, and at other desired points inside and outside the Residence. Each thermocouple was connected through switches to an indicating potentiometer. It was possible to obtain a continuous record of any 28 of the 180 thermocouple stations by using recording potentiometers. Resistance thermometers were installed at the sitting level on the standards in the first-story rooms. These resistance thermometers were connected to a six-point recording resistance bridge which provided a continuous record of the temperatures at these selected points.

Heat-flow meters were installed in various locations in the house. The four meters which were installed in the outside walls were located on the exterior

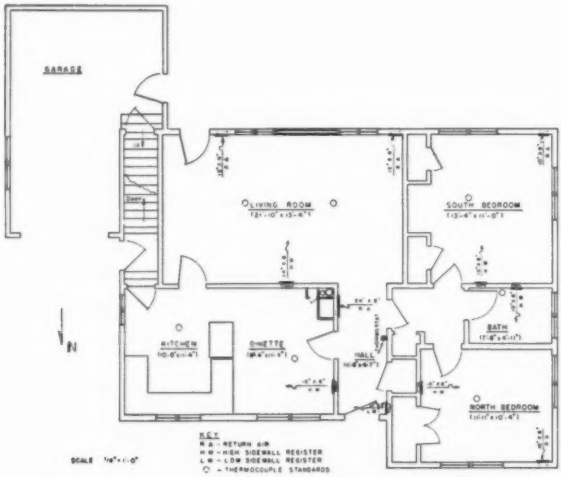


FIG. 3. FLOOR PLAN OF RESEARCH RESIDENCE NO. 2

surface of the interior panels and next to the insulation. One such meter was located in each wall, 60 in. from the floor. Those which were placed above the ceiling were between the top ceiling surface and the attic insulation. One meter was also placed on the underside of both the north and south slopes of the roof.

Continuous records were made of the outdoor-air temperature, both dry bulb and wet bulb, the indoor relative humidity, the wind velocity, and the intensity of solar radiation received on a horizontal surface. The wind direction was noted when all observations were made.

Experimental Conditions: One main series of studies was conducted, which provided for 120 cfm of ventilation air (one air change per hr) to be introduced mechanically into the return-air side of the system. In this series the total air-flow rate through the unit was approximately 600 cfm, or 300 cfm per ton of rated capacity of the refrigeration unit. The blower was operated continuously, independent of the operation of the condensing unit. The indoor dry bulb control

temperature was 75 F, but no attempt was made to control the indoor relative humidity.

The house was operated with all exterior doors and windows closed at all times. The first story was also separated from the basement by a door at the top of the basement stairs. Although all windows were equipped with venetian blinds, they were raised to the full open position and the windows were left completely uncovered except for the awnings and the roof overhang mentioned previously in the description of the residence.

Procedure: Room-air, outdoor, interior and exterior wall-surface temperatures, and temperatures within the wall were recorded at least twice daily, in the morning before the sun imposed a load on the house, and in the latter part of the afternoon when the peak load occurred. Power consumption, operation time, and water consumption by the unit were recorded at 5:30 a.m. CST each morning, the time selected to start the 24-hr test day.

Special studies of from 48 to 72 hr duration were made periodically. These studies were made during periods of clear hot weather and were made for the purpose of determining the time at which maximum cooling load was imposed on the residence, and in turn on the condensing unit, the time lag of the heat flow through the walls, and the effect of continued hot weather on the cooling load. Temperature and heat-flow meter observations were recorded hourly for approximately 18 hr each day during the special-study periods.

RESULTS

Determination of Actual Cooling Load: The actual cooling load for the residence was determined by making a heat-balance of the various measured quantities at the condensing unit. The heat balance consisted of three components:

1. The heat equivalent of the power input to the compressor.
2. The heat removed from the conditioned air in the evaporator.
3. The total heat absorbed by the cooling water in the condensing unit.

It was assumed that the heat absorbed by the cooling water while passing through the condensing unit could be considered as the sum of the heat absorbed by the refrigerant in the evaporator and the heat equivalent of the power input to the compressor. The two factors which validate this assumption were previously mentioned in the description of the equipment, *i.e.*, the motor casing was water cooled and the condensing unit housing was insulated.

The data used to determine the actual cooling load imposed on the residence were abstracted from the results of a special study for a typical day. The day selected was July 22, 1952, because the hourly outdoor-air temperatures and the daily range of outdoor-air temperature closely approximated those of a design day. Values abstracted from hourly observations were water-flow rate, power input to the compressor and water temperature rise through the condenser. The method used in making the heat balance can best be illustrated by an example, and for this purpose the values observed at 3:00 p.m. CST, the time of maximum calculated load were employed. The heat balance was completed by applying the following equation:

$$H_e = H_e + H_{pi}$$

where

H_c = heat absorbed by the cooling water from the point of entry into the unit housing to the point at which it leaves the unit housing, Btu per hr.
H_e = heat absorbed from the air passing through the evaporator. This heat represents the net heat gain to the house including gains from walls, glass, ceiling, and ventilation air but excludes any loss through the basement walls, Btu per hr.
H_{pi} = heat equivalent of the power input to the compressor, Btu per hr.

The cooling load, H_e, is then equal to the difference $H_c - H_{pi}$.
For any experimental hour

$$H_c = 8.33\, QC_p\, (t_o - t_i)$$

where

Q = water-flow rate, gal per hour.
C_p = specific heat of water Btu per (pound) (Fahrenheit).
8.33 = density of standard water, pound per gallon, and
$(t_o - t_i)$ = temperature difference between water outlet and inlet through condensing unit, Fahrenheit.

For the time under consideration

$$H_c = 8.33 \times 136\ (23.5) = 26{,}600 \text{ Btu per hour.}$$

The heat equivalent of the power input to the unit can be found at any time as

H_{pi} = 3,413 × kilowatt-hour, where 3,413 is the heat equivalent of electrical energy, Btu per kilowatt-hour.

This value of H_{pi} for the hour under consideration is

$$H_{pi} = 3{,}413 \times 1.8 = 6{,}150 \text{ Btu per hour.}$$

The cooling load for this hour was then found to be (26,600—6,150) Btu per hr or 20,450 Btu per hour. The cooling load was calculated by this method for each hour of every cycle throughout the day. Fig. 4 shows these values as a function of time. Since the capacity of the unit varied from hour to hour with changes in return-air conditions, the hourly values of capacity have been plotted at the beginning and the end of each cycle and connected with straight lines. This has been done because the actual variation in load from point to point cannot be readily determined.

The actual cooling load can also be determined as the heat removed from the air in the evaporator coil. The heat removed by the evaporator coil was computed in two parts: the sensible heat removed from the air, and the latent heat absorbed in condensing the moisture from the air. The sensible heat removed from the air at any time can be calculated as:

$$H_s = Q \times d_a \times c_p \times \Delta t$$

where

H_s = sensible heat removed from the air, Btu per hour.
Q = air-flow rate, cubic feet per hour.
d_a = density of standard air, 0.075 lb per cubic foot.
c_p = specific heat of air at constant pressure, 0.24 Btu per pound, Fahrenheit.
Δt = temperature drop through the evaporator coil, Fahrenheit.

The latent heat removed at any time can be calculated as:

$$H_L = 1080\ W_c$$

where

H_L = latent heat removed from the air, Btu per hour.
W_c = moisture condensing rate, pound water per hour.
1080 = the approximate amount of heat removed from the condensed moisture including superheat, latent heat, and sub-cooling, Btu per hour.

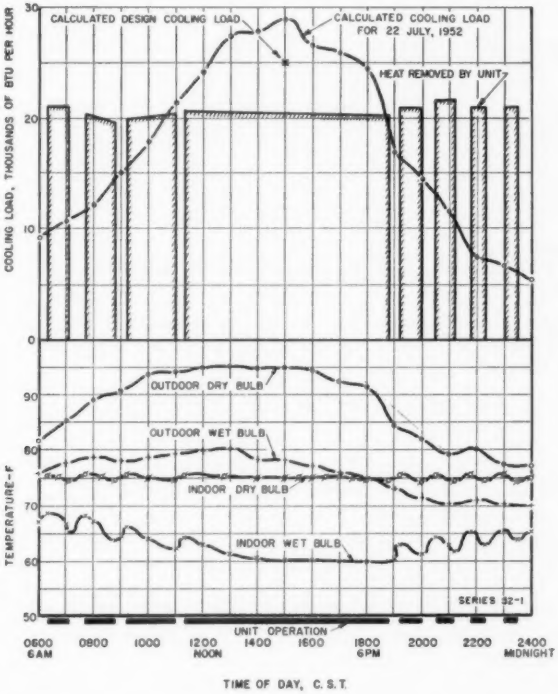


FIG. 4. ACTUAL AND CALCULATED COOLING LOADS OF RESIDENCE FOR JULY 22, 1952

The preceding calculations have been made for the typical hour selected with the following results:

H_s = 37,320 × 0.075 × 0.24 × 22.4 = 15,080 Btu per hour.
H_L = 1,080 × 4.9 = 5,300 Btu per hour.

The sum of these two values gives a total of 20,380 Btu per hour as the total heat absorbed in the evaporator. This is a deviation of —0.34 percent from the

TABLE 3—TYPICAL OPERATING DATA AND RESULTS—3:00 P.M. CST, JULY 22, 1952

1. Outdoor Air Condition	94.8 DB, 78.1 WB, 48% RH
Humidity Ratio, lb water per lb dry air	0.01700
Wind	9 mph SW
2. Indoor Air Condition	
(Sitting Level Temp; Avg of all Rooms)	76.1 DB, 60.6 WB, 41% RH
Humidity Ratio, lb water per lb dry air	0.00780
3. Ventilation Air Condition	89.0 DB, 76.7 WB, 57% RH
Humidity Ratio, lb water per lb dry air	0.01700
4. Condition of Mixed Air Entering Coil	77.3 DB, 64.0 WB, 48% RH
Humidity Ratio, lb water per lb dry air	0.00967
5. Condition of Air Leaving Coil	54.9 DB, 52.1 WB, 83% RH
Humidity Ratio, lb water per lb dry air	0.00760
6. Air Temp Drop Through Cooling Coil, F	22.4
7. Temp of Cooled Air Leaving Registers (Avg of all Registers) F	60.4
8. Air Temp Rise in Ducts and Casing, F	5.5
9. Basement-Air Temp at Sitting Level (Avg of all Stations) F	77.2
10. Flow Rate of Air Circulated, cfm	622
11. Flow Rate of Ventilating Air, cfm	126
12. Moisture Condensed from Air, lb per hr	4.9
13. Heat Absorbed by Cooling Coil, Total, Btu per hr	20,380
Heat Due to Moisture in Air, Btu per hr	5,300
Sensible Heat, Btu per hr	15,080
14. Water Temp Through Condensing Unit, F	Inlet 62.2, Outlet 85.7; rise 23.5
15. Flow Rate of Condenser Water, gal per hr	136
lb per hr	1133
16. Heat Absorbed by Water Passing Through Condensing Unit, Btu per hr	26,600
17. Heat Equivalent of Power Input to Compressor Motor, Btu per hr	6,150
18. Net Cooling Load (Item 16—Item 17), Btu per hr	20,450

net cooling load as calculated previously by using the heat balance around the condenser. During periods of continuous operation the deviation of actual cooling loads as determined by the two methods varied by less than ±5 percent which indicates a good agreement between the two methods. For short cycles of operation, however, the heat balance at the condensing unit must be relied upon to determine the cooling load since unsteady conditions at the evaporator make an energy balance too difficult to be of any practical use. Table 3 shows the pertinent operation data and results for the typical hour selected. All of the values tabulated are measured data or have been computed by using measured values as the basis for the computation.

Time Lag: Although the cooling load imposed by the heat flow through the walls and ceiling was small in comparison to that imposed through the windows, the time lag between the outer-surface and the inner-surface maximum temperatures was important in fixing the time of the maximum cooling load. If the time lag is to be defined in this manner, however, it must be recognized that factors other than the exterior-surface temperature and the room-air temperature can affect the temperature at the inner surface of the wall or ceiling. It can be seen in Figs. 5 and 6 and in Table 4 that in several cases the interior-surface temperature attained a maximum value before either the temperature on the room-side of the insulation or the heat flow reached a maximum. A possible explanation of this may be the combined effects of solar radiation and outdoor temperature entering the rooms through the relatively large window areas. This

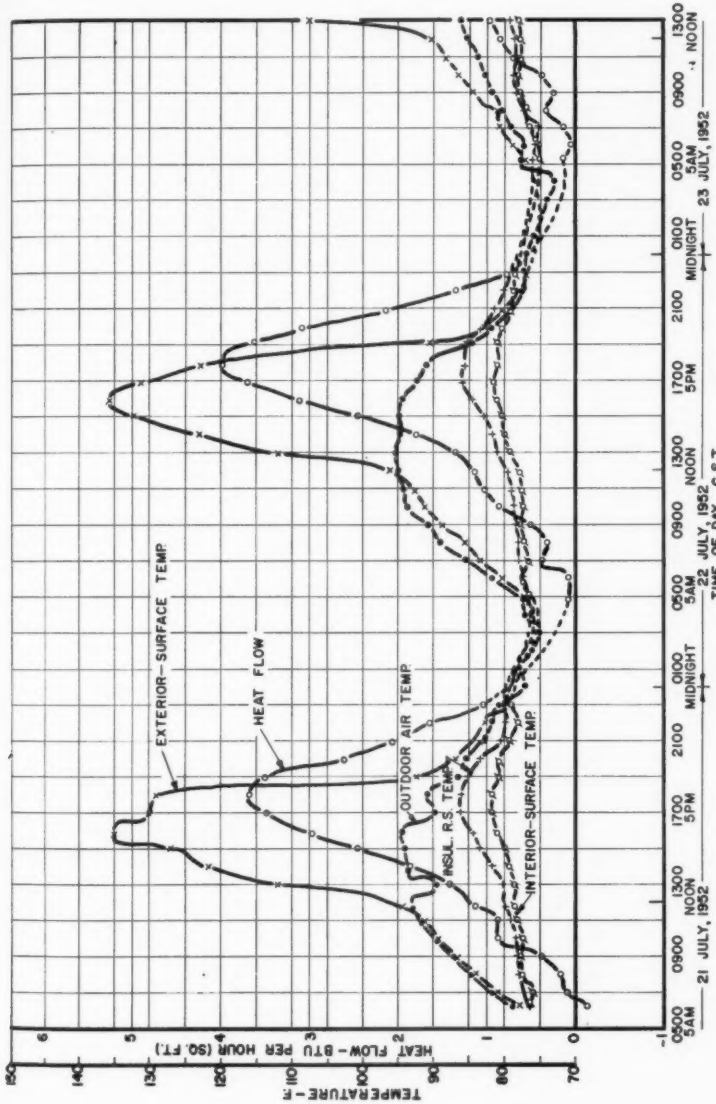


FIG. 5. SURFACE TEMPERATURES AND HEAT FLOW THROUGH WEST WALL

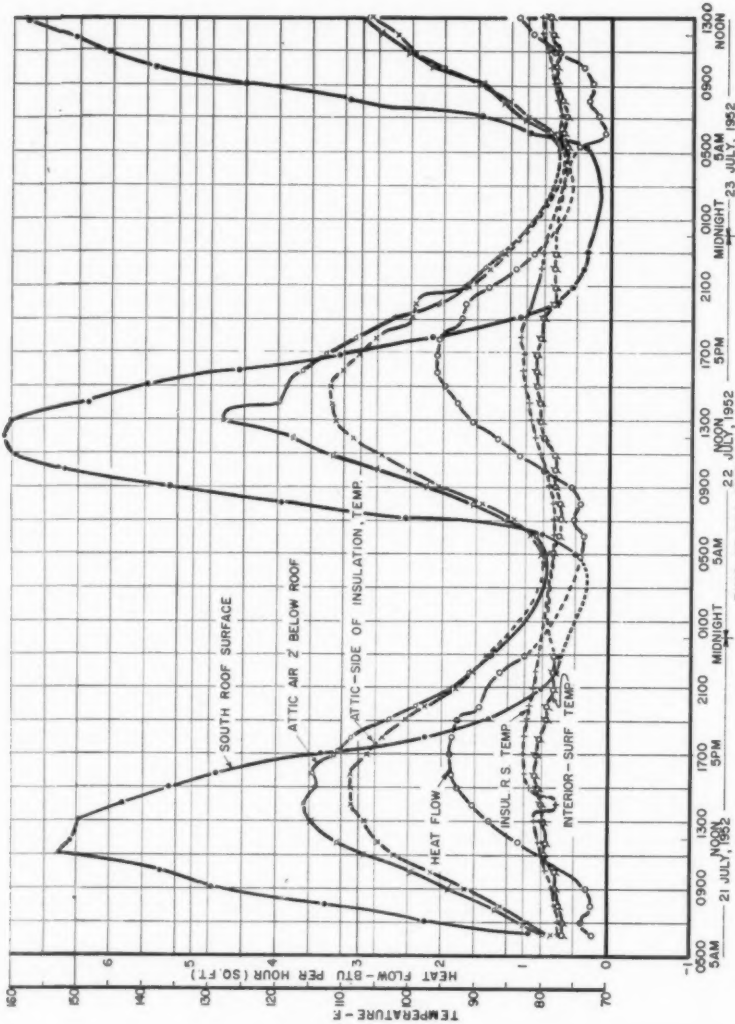


FIG. 6. SURFACE TEMPERATURES AND HEAT FLOW THROUGH LIVING ROOM CEILING

TABLE 4—TIME LAGS AND TIMES OF OCCURRENCE OF MAXIMUM TEMPERATURES AND HEAT FLOW IN THE WALLS AND CEILINGS OF RESEARCH RESIDENCE NO. 2

DATE	WALL ORIENTATION	TIME MAXIMUM VALUE OCCURRED, CST				EXTERIOR SURFACE TO INTERIOR SURFACE TIME LAG, HR COL (4)-COL (1)
		Exterior Surface Temp	Room Side of Insulation Temp	Heat Meter	Interior Surface Temp	
		(1)	(2)	(3)	(4)	
7-21-52	East	9:00 a.m.	11:10 a.m.	11:15 a.m.	11:00 a.m.	2.0
	South	2:00 p.m.	4:15 p.m.	3:20 p.m.	3:50 p.m.	1.8
	West[a]	4:10 p.m.	5:40 p.m.	6:10 p.m.	5:30 p.m.	1.3
	North	6:00 p.m.	6:10 p.m.	7:10 p.m.	6:15 p.m.	0.3
	Ceiling[a]	3:00 p.m.	5:00 p.m.	5:00 p.m.	4:25 p.m.	1.4
7-22-52	East	9:20 a.m.	11:50 a.m.	11:00 a.m.	12:00 noon	2.7
	South	1:20 p.m.	3:30 p.m.	2:40 p.m.	2:30 p.m.	1.3
	West[a]	4:10 p.m.	5:40 p.m.	5:50 p.m.	5:25 p.m.	1.3
	North	5:45 p.m.	6:45 p.m.	8:00 p.m.	6:30 p.m.	0.8
	Ceiling[a]	2:40 p.m.	5:00 p.m.	4:30 p.m.	4:15 p.m.	1.6

[a] Values for these orientations may be noted from Figs. 5 and 6, respectively.

energy which entered the structure by radiation through the windows undoubtedly caused an increase in the temperature of the interior surface by re-radiation from the floor and other wall surfaces. Large window areas such as were used in the residence are commonly found in residential construction, and thus the effects of diffuse and direct solar radiation are more severe in cooling this type of structure.

The time lags through the walls and ceiling did not remain constant from day to day. Table 4 shows that only the time lag on the west wall of the residence remained constant from July 21 and 22. Figs. 5 and 6 show that the maximum temperatures measured within the west wall and the ceiling did not change more than 0.5 F from July 21 to 22, and that the maximum heat flow changed less than 10 percent. Similar measurements on the other walls of the residence showed the same results. Furthermore, during the two days preceding this study, the maximum outdoor-air temperature was 90 F and above and the sky was mostly clear. The variations in time lag therefore cannot be attributed to a temperature build-up within the walls and ceiling. That this is true would also be indicated by the inconsistent variations in the various walls. A combination of indoor and outdoor factors such as air-temperature, humidities, wind, and sun must have been causing this effect.

Despite the lack of exact information on the time lag of the heat flow imposed by the outdoor-air temperature and solar radiation, it is apparent that it was much less than the lag which would be estimated from information in THE GUIDE 1952. The estimated lag for the type of construction in the Residence was 5 hr; the measured lag was less than 3 hr in all cases. Although it has been shown that the interior-surface temperature and consequently the apparent time lags are affected by factors other than the exterior-surface temperatures, the heat transfer from the interior surface of the room to the room air is a function of the interior-surface temperature, the film coefficient along the surface, and

the room-air temperatures. Thus, the factors which affect the surface temperatures and the time at which their maximum values occur are important only insofar as they facilitate prediction of the maximum values and the time of their occurrence. The design procedure now in use does not predict the time of occurrence of the maximum interior-surface temperatures. The magnitude of the maximum interior-surface temperatures will be discussed later in terms of heat flow.

Figs. 7, 8, and 9 show comparisons between the calculated and measured heat flows through two walls and the ceiling of the residence. Because each heat-

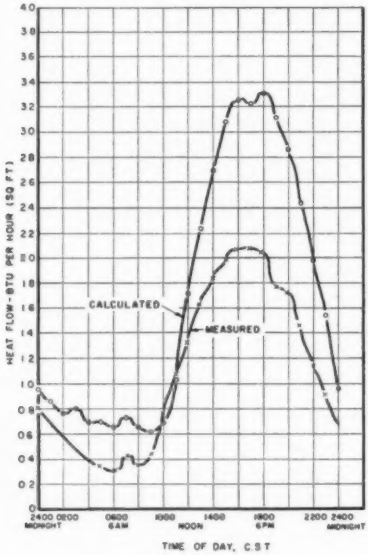


FIG. 7. CALCULATED AND MEASURED HEAT FLOW THROUGH LIVING ROOM CEILING

flow meter was placed between the insulation and the interior wall or ceiling panel and midway between studs or joists, the calculated heat gain is that for an insulated section of the wall or ceiling. Thus, it is based on an overall coefficient of 0.06 Btu per hr (sq ft) (F) used for the wall section rather than the value of 0.07 Btu per hr (sq ft) (F) used in the cooling load calculations.

Comparisons of the measured and the calculated values of the heat flow through the walls and ceiling of the residence show that in general the measured value was less than the calculated value when the wall was not exposed to direct solar radiation. This is shown in Fig. 7 in which the curve of measured heat flow is below the calculated curve except for a two-hour period in the morning. This discrepancy was attributed to the difference between the measured and

predicted time lags. The same general conclusions were drawn from studies of the heat flow through the north and east walls of the residence. It should be noted here that the ceiling-attic-roof combination was treated as a flat roof in calculating the cooling load because the design procedure applied only to horizontal flat roofs.

The measured heat flow through the west wall was considerably greater than the calculated heat flow as shown in Fig. 8. The discrepancy was greatest during periods of greatest solar load on the wall. Although it would appear that the measured heat flow became greater than that calculated at 9:00 a.m. before the

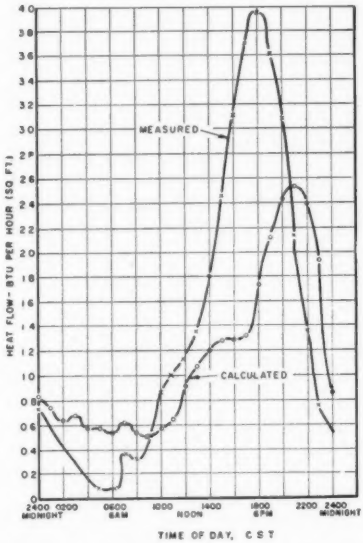


FIG. 8. CALCULATED AND MEASURED HEAT FLOW THROUGH WEST WALL

sun imposed direct radiation on the wall, the difference between the actual and predicted time lags indicated that if the calculated load were shifted three hours to the left, the curves would cross at approximately 1:00 p.m.; this would be shortly after the time at which the sun could begin to radiate directly to the west wall. That the maximum measured flow [3.95 Btu per hr (sq ft)] was 50 percent greater than the maximum calculated heat flow [2.53 Btu per hr (sq ft)] can be attributed to the direct solar radiation on the west wall of the residence and apparent discrepancies between the actual sol-air temperatures and those tabulated in THE GUIDE 1952.

The measured heat flow through the south wall of the living room was less than the calculated heat flow until after 5:00 p.m. This is shown in Fig. 9 which does not make allowance for the difference between the actual and the predicted time lags. The calculated heat flow was that for an unshaded south

wall having a U value of 0.06 Btu per hr (sq ft) (F). Although the discrepancy between the maximum calculated and the maximum actual heat flows was less than 20 percent, it is obvious that the measured heat flow would have been considerably greater had the wall not been shaded by the overhang of the eave. As a consequence it is somewhat doubtful if the calculated load would be greater than that measured on an unshaded south wall.

Actual and Calculated Cooling Load Compared: Before attempting to draw any comparison of the actual and calculated cooling loads the significant assumptions necessary to make these calculations for the residence must be stated. As

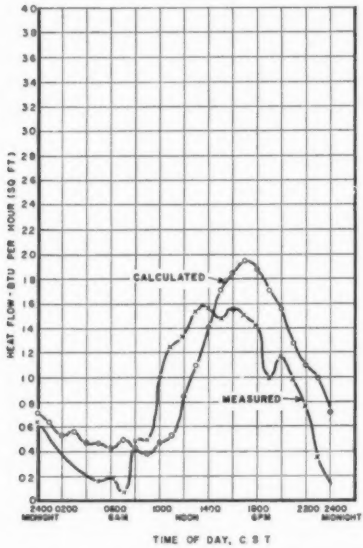


FIG. 9. CALCULATED AND MEASURED HEAT FLOW THROUGH SOUTH WALL

mentioned previously, the sol-air temperature method was used to calculate the heat flow through the walls and ceiling, and for July 22 the day selected, the design sol-air temperatures were adjusted for the outdoor-air temperatures experienced in accordance with the correction factors given in THE GUIDE 1952. The roof-attic-ceiling combination was considered as a flat roof as mentioned previously. The attic of the residence was considered to be well ventilated since louvers were installed in the east and west gables and continuous openings under the eaves 2 in. wide extended along both north and south edges of the residence. For this reason, the sol-air temperatures for the roof surface could not reasonably be applied to the attic side of the ceiling insulation. For these calculations the surface absorptivity for solar radiation was assumed to be 0.7 rather than the 0.9 value suggested in THE GUIDE 1952 for dark roof surfaces. In effect,

this value of 0.7 lowered the sol-air temperatures that were applied at the attic side of the insulation approximately 22 percent.

The shaded south wall of the residence was considered to receive direct radiation from the sun when computing the wall heat gains for lack of information regarding a shaded wall. The sol-air temperatures for a north wall could be considered to apply to shaded walls; however the differences in the values of sky radiation for the two exposures and in the times of peak temperature indicated this would introduce an error. The error introduced by considering the south wall to be unshaded would be on the conservative side but could become serious if the south wall contained a very small glass area. The heat gain through the walls which did not receive direct solar radiation such as the east wall in the north bedroom (see Fig. 3) and the east wall of the living room, was considered a function of indoor-outdoor temperature difference only taking into account the predicted 5-hr time lag.

The heat gain through the windows was computed from the tables presented in THE GUIDE 1952. A shading factor of 0.3 was applied to the solar heat gain for all windows equipped with awnings, and for all windows on the north and

TABLE 5—COMPARISON OF MEASURED AND CALCULATED HEAT REMOVAL BY UNIT FOR JULY 22, 1952

PERIOD OF DAY	6:00 a.m.-11:00 a.m.	11:00 a.m.-7:00 p.m.	7:00 p.m.-MIDNIGHT	TOTAL
Measured	73,900	153,500	53,700	281,100
Calculated	71,200	204,000	51,500	326,700
Percentage Difference (Based on Measured Value)	−3.7	+32.9	−4.1	+16.2

south exposures which were shaded by eave overhangs. No attempt was made to correct the heat gain through the windows for deviations of indoor-outdoor temperature difference from those on which the tables are based.

Although the calculated design cooling load of the residence was 25,000 Btu per hour, the actual heat removed from the residence did not exceed 22,000 Btu per hour when the outdoor design conditions were reached or exceeded. Fig. 4 shows the hours of operation and amount of heat removed for July 22, 1952, a day on which the maximum outdoor dry bulb temperature was 95 F and the maximum outdoor wet bulb temperature was 80 F, 4 deg greater than the design wet bulb temperature of 76 F. The average wind velocity for the day was 6 mph from the southwest. The cooling load for this day was calculated by using the design indoor conditions (75 F dry bulb, 62.5 wet bulb) and the actual outdoor conditions to adjust the design sol-air temperatures in accordance with the correction factors given in THE GUIDE for computing the heat flow through the walls and ceiling. The heat flow through the ceiling was calculated by the method outlined in Chapter 12 of THE GUIDE 1952 (p. 257) on the Cooling Load but by using the adjusted sol-air temperatures. The heat gain through the windows was calculated by using the same values as were used in the design calculations.

The actual heat removed from the residence during the 18 hr period shown is less than the calculated heat removed would indicate. Integration of the areas

under the curves of heat removed by the cooling unit shows the total to be 281,000 Btu for the eighteen-hour period. The predicted heat removal, obtained by integrating the area under the calculated cooling load curve, was 326,700 Btu, 16 percent greater than that actually removed.

A breakdown of the comparison into periods of the day is of more significance than comparing the total heat removal. Table 5 shows this comparison. For convenience, the test period has been divided into three parts: (1) the morning during the period when the cooling load was increasing; (2) the period of maximum cooling load during the hottest part of the day; and (3) the period when the cooling load was decreasing during the evening. It should also be noted that these periods coincide with the time of intermittent operation during the morning, continuous operation during the afternoon, and intermittent operation during the evening.

Between the beginning of the study at 6:00 a.m. and the end of the last of the intermittent operations at 11:00 a.m., the unit removed 73,900 Btu from the air passing through the evaporator coil, 2,700 Btu more than would be predicted from the area under the calculated curve. Although this difference is small, less than 4 percent, it is felt that during this period there should have been an increase in the internal energy of the materials and furnishings within the residence.

This heat storage effect is shown during the period of continuous unit operation from 11:00 a.m. to 7:00 p.m. The energy removed by the unit during this long operation was 153,000 Btu, more than 32 percent less than the value of 204,000 Btu predicted by calculation. Much of this difference can be attributed to energy entering the exterior surfaces of the residence and being stored within the structure, but the magnitude of heat storage is not known. That heat storage was at least partially responsible for the difference is evidenced by two factors: (1) the constant capacity of the cooling unit, and (2) the constant air temperature at the thermostat. Even though the average room-air temperature at the sitting level rose slightly more than 1 deg during the afternoon, the energy stored in the air within the residence due to this temperature rise was insignificant.

The energy actually removed by the cooling unit during the evening (between 7:00 p.m. and midnight) was more than 4 percent greater than the calculated value of 51,500 Btu. This again shows the heat storage effect of the residence; during the period of decreasing load, the amount of energy predicted to enter the house was less than the amount actually removed. The difference between the measured and the calculated cooling loads was much smaller than would be expected, since it should be during this period that the greatest reduction in heat storage would occur.

Operating Characteristics of the Unit: As indicated in Fig. 4, the cyclical variation in room-air temperature (indoor dry bulb) was small. This small variation was characteristic of the conditions which were maintained in all of the first-story rooms during the cooling season, even though the indoor dry bulb temperature shown in Fig. 4 was measured at the room thermostat. When intermittent operation of the unit occurred, the variation in room-air temperature amounted to about 1 deg between cycles of operation. However, when the unit operated for prolonged periods of time, there was no appreciable variation in room-air temperature experienced.

The difference in room-air temperature between rooms in the Residence and

at the thermostat was slightly greater than 1 deg as a maximum depending upon the time of day and the outdoor weather conditions experienced. For instance, during periods of peak load the room-air temperatures at the sitting level in the south bedroom and in the kitchen-dinette were greater than those experienced in the other rooms of the residence. The higher room-air temperature in the south bedroom was attributed to the relatively large cooling load imposed upon this room through the ceiling and through the south and west exposures as compared with the air supply to it through the one high sidewall register, and that in the kitchen to the fact that the air was delivered to the kitchen-dinette through one register located in the west wall of the dinette. The room-air temperature in the bathroom and in the north bedroom, on the other hand, was lower than that at the thermostat. This was attributed to these rooms being smaller than the south bedroom and the air supply being more adequate as compared to the load imposed upon these rooms.

The compressor operated a total of 419 hours during the period from June 7, 1952, to July 24, 1952. The total number of degree-hours above 75 F (outdoors) was 6,154 or an average of 131 per day for the period. The maximum for one day was 318 degree-hours above 75 F and the minimum was zero.

The water-flow rate through the condenser varied from 135 to 140 gal per hour of operation. Based upon the measured capacity of the condensing unit the flow rate was approximately 1.35 gal per min per ton. The temperature of the water entering the condenser increased from 62 F at the beginning of the cooling season to 65 F at the end of the season, and the water temperature rise through the condenser varied from 22 to 25 deg depending upon the length of operation. The total water consumption for the period June 7, 1952, to July 24, 1952, was 57,700 gallons (7720 cu ft). The compressor consumed 1.8 to 1.9 kw or 1.1 kw per measured ton, and the power consumption for the blower motor was approximately 0.2 kw. For the same period mentioned above, June 7, 1952, to July 24, 1952, the total power consumption was 836 kwhr by the compressor and 229 kwhr by the blower motor.

The compressor operated a maximum of 16.2 hr for one day during the same period. This was also the maximum operating time experienced during the cooling season. The water and power consumptions of the unit on the day of maximum operation was 2260 gallons and 30.6 kwhr, respectively. On an average day during this period, the unit operated 8.72 hours. The water and power consumption of the unit for the average day were 1216 gallons and 17.5 kwhr, respectively. During any day of the period, the blower motor consumed 4.9 kwhr.

SUMMARY

The air conditioning system discussed in this paper consisted of a year-around air conditioner which contained a two-ton condensing unit and was connected to a duct system which was designed only for heating the residence in the winter. The registers were located in the high side-wall location on the interior walls of the rooms. The total air-flow rate through the unit was approximately 600 cfm of which 120 cfm was introduced mechanically from outdoors, and the blower was operated continuously.

The indoor dry bulb control temperature was 75 F, but no attempt was made to control the indoor relative humidity. The east and west windows of the

residence were equipped with canvas awnings and the south windows were shaded by the overhanging eave. The windows were not otherwise covered. The residence was unoccupied and all windows and exterior doors were closed at all times; the first-story rooms were separated from the basement by an interior door at the top of the basement stairs.

Cooling Load: On a day during which the hourly outdoor-air temperatures closely approximated those of a design day, the maximum measured cooling load was 20,450 Btu per hour. This value is considerably lower than the calculated design cooling load of 25,000 Btu per hour which was obtained using the sol-air temperature method given in THE GUIDE 1952. The cooling load calculated by this method for the day previously mentioned was greater than the design cooling load because the outdoor wet bulb temperature exceeded that assumed for a design day.

Heat Flow: The time lag of the heat flow through the wall and ceiling of the residence was found to be from 2 to 3 hrs. The time lag predicted from THE GUIDE 1952 was 5 hr. It was also found that in several cases the temperatures of the interior surface of the exterior walls reached their maximum values before the temperatures within the walls reached their maximum values. Little consistency was noted, however, in the time lags through the various walls and they changed from day to day. No evidence of temperature buildups within the exterior walls was noticed.

The measured heat flow through those walls which were exposed to direct solar radiation was found to be higher than that which was calculated. In the case of the south wall and the ceiling, however, the calculated heat flow was found to be greater than the measured heat flow. The discrepancy between the measured and calculated heat flow in the south wall may be attributed to the assumption of an unshaded wall used in the calculations. In general, however, the measured heat flow was less than the calculated value for surfaces which were not exposed to direct solar radiation and greater than the calculated value for surfaces which were exposed to direct solar radiation.

Total Heat Removed: The total heat removed during the 18-hour period from 6:00 a.m. to midnight on July 22, 1952, was 16 percent less than that which would be predicted from the calculated cooling load for that day. The greatest discrepancy occurred between 11:00 a.m. and 7:00 p.m., the period during which the compressor operated continuously; the calculated heat removal was almost one-third greater than that which actually occurred. In the morning and evening, however, the calculated heat removal was slightly less than the measured value.

Operating Costs: The unit operated a total of 419 hours during the period from June 7, 1952, to July 24, 1952. During this same period, the water consumed by the unit was 57,700 gallons (7,720 cu ft). The power consumption by the compressor and the blower motor throughout this same period was 836 kwhr and 229 kwhr, respectively.

ACKNOWLEDGMENTS

This paper reports one phase of a comprehensive investigation conducted in Warm-Air Heating Research Residence No. 2 under the terms of a cooperative agreement between the *National Warm Air Heating and Air Conditioning Association* and the *Engineering Experiment Station of the University of Illinois.*

These results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

Acknowledgment is made to the manufacturers who cooperated by furnishing equipment used in the investigation.

REFERENCES

1. Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 95).
2. Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 167).
3. Study of Unit Room Coolers in the Research Residence, by A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 77).
4. Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207).
5. Study of Summer Cooling in the Research Residence Using Water from the City Mains, by A. P. Kratz, M. K. Fahnestock, S. Konzo and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 193).
6. Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 287).
7. Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, M. K. Fahnestock, S. Konzo and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 439).
8. Study of Summer Cooling in the Research Residence for the Summer of 1938, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 591).
9. Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P. Kratz, S. Konzo and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 203).
10. HEATING VENTILATING AIR CONDITIONING GUIDE 1952 (published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.).
11. Comparative Performances of a Warm-Air Ceiling Panel System and a Convection System, by R. W. Roose, M. E. Childs, G. H. Green and S. Konzo (University of Illinois *Engineering Experiment Station Bulletin 401*, June 1952).
12. Manual 7, Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (*National Warm Air Heating and Air Conditioning Association*, 2nd edn., 1947).

DISCUSSION

G. V. PARMELEE, Cleveland, Ohio: The work reported by Mr. Gilkey and his associates at the University of Illinois is of great value for two reasons.

1. It puts new emphasis on the need for understanding more thoroughly the role played by the heat capacity of the internal structure of a building in causing the actual cooling load to differ from the instantaneous rate of heat gain. Practical steps toward understanding and estimating the magnitude of this effect are part of the current A.S.H.V.E. research program.

2. It represents, I believe, one of the first attempts to correlate actual field test data with the periodic heat flow theory, since Professor Mackey in the early 1940's brought out his very useful sol-air temperature concept. This concept provided a

practical, though perhaps not yet much used, method of calculating heat gains during a periodic variation in air temperature and solar radiation incident upon a wall or a roof.

I would like to see the authors go a bit farther than they have in comparing the measured heat flow rates with calculated heat flow rates. Their calculated values have been based upon the design values of sol-air temperature given in THE GUIDE 1952. The big difference between calculations and results suggests two things: (1) that perhaps our design values of sol-air temperature are not comparable with those found in practice (this suggests that we ought to restudy the problem), and (2) that perhaps the theory itself is not completely compatible with practice. I would like to suggest that, since the authors have the data on the incident solar radiation on a horizontal plane, they make a new estimate of the sol-air temperatures that prevailed during the test. From these values the heat gains could be calculated and compared with the observed heat gains. This might give us a clue indicating whether unsuitable design data or inadequacies of the mathematical theory are at fault.

I. A. NAMAN, Houston, Tex.: I would like to know whether the ventilation air was measured constantly to determine whether or not infiltration was affecting the rate of flow of the introduction of outside air and whether or not wind velocities were taken into account in trying to correlate this data with the calculated values.

AUTHORS' CLOSURE (H. T. Gilkey): The ventilation air-flow rate was measured constantly and it did not vary appreciably with wind direction and velocity in the range experienced. No attempt has been made to correlate the conductance at the exterior surface of the wall with the wind velocity.

Considerable data on the incident solar radiation on a horizontal plane have been collected, but at this time they have not been analyzed. It is hoped that we shall be able to do this in the near future. As you probably know, the sol-air temperature concept was devised under conditions of no energy entering the enclosure by means of radiation through windows, etc. However, most homes have windows and it is our feeling that the design methods should be revised to take this into account. It seems that they should also be revised to make them more usable from a practical standpoint.

1478

No. 1478

SEMI-ANNUAL MEETING, 1953

DENVER, COLORADO

With a total attendance of 618, the 1953 Semi-Annual Meeting of the Society set a new record for summer meetings. The Committee on Arrangements of the Rocky Mountain Chapter provided a full round of activities for the 273 members, 234 ladies and children, and 111 guests who attended the meeting and sessions at the Shirley-Savoy Hotel in Denver, June 29-July 1.

The first session with Pres. Reg. F. Taylor presiding was called to order at 9:30 a.m. on Monday, June 29th in the Empire Room at the Shirley-Savoy Hotel. President Taylor called on Fred Janssen, general chairman of the Committee on Arrangements, who welcomed the members to the meeting and briefly outlined the program of special events planned.

The Society's loss in the untimely death of Cyril Tasker, Director of Research, was mentioned by President Taylor and he called on First Vice Pres. L. N. Hunter to present the following resolution, which was prepared by Mr. Hunter, I. W. Cotton and A. W. Edwards.

The sudden and untimely decease of Cyril Tasker on May 27, 1953, brought a poignant loss to the A.S.H.V.E. and to our profession.

Mr. Tasker was a loyal active member of the Society for many years, having been president of Ontario Chapter and a member of the Council.

His service as Director of Research began in 1943. With the removal of the Laboratory from Pittsburgh to Cleveland, he recruited a new staff and built an effective organization which functioned in new quarters as a dynamic activity. The loyalty and affection of his associates testify eloquently to his capacity as an administrator.

A tireless worker, he carried out his responsibilities as Director of Research conscientiously and effectively.

The engineering profession recognized Mr. Tasker as an international authority in his field. He wrote and spoke convincingly, with refreshing absence of stilted professionalism.

The Council of A.S.H.V.E., recognizing the great loss in his passing, directs that this expression of sadness and appreciation be spread on Council minutes and published as representing the sentiments of the Society; and further directs that a copy be sent to Mrs. Tasker as an expression of sympathy from the entire Society membership.

President Taylor also announced with regret the passing of Past Pres. F. E. Giesecke in New Braunfels, Tex., on June 27, and appointed a committee composed of D. M. Mills, L. N. Hunter and John W. James to prepare a suitable resolution with reference to Dr. Giesecke.

In tribute to Dr. Giesecke and Mr. Tasker, the audience rose and stood in silence for one minute.

President Taylor appointed a Committee on Resolutions to report at the final session composed of M. W. Bishop, Chicago, Ill., *chairman;* W. J. Collins, Jr., Oklahoma City, Okla., and F. H. Faust, Bloomfield, N. J.

The Charter and By-Laws Committee had prepared amendments to the By-Laws based upon the recommendations of the Council. These were presented by Mr. Hunter in the form of a resolution as follows:

ARTICLE VIII—Meetings, Nominations and Elections

Section 7. Results

. . . The elected candidates shall be installed, and their terms shall commence, at the close of the last session of the Annual Meeting.

To be amended as follows:

. . . The elected candidates shall be installed during the Annual Meeting and their terms shall commence at the close of the last session of the Annual Meeting.

ARTICLE VII—Committees

Section 2. Council Committees

(e) Standards Committee, which shall consider all scientific questions and data pertaining to engineering codes and standards, initiate and propose changes and improvements thereof for the public benefit and report its recommendations to the Council.

This paragraph to be deleted and the following paragraph inserted.

Section 3. General Committees

(j) Standards Committee, consisting of six (6) Members. The said committee shall consider all scientific questions and data pertaining to engineering codes and standards, initiate, and propose changes and improvements thereof for the public benefit, and report its recommendations to the Council.

It was moved by Mr. Hunter, seconded and voted unanimously that the resolution be adopted and the proposed amendments would be submitted to a vote of the Society at the Annual Meeting in 1954.

There were three technical papers presented at the first session at which John W. James served as Chairman. (See program, page 308)

At the second session called to order on Tuesday, June 30th at 9:30 a.m., Pres. Reg. F. Taylor presided and W. F. Friend was Chairman. Three technical papers were presented and discussed.

The third technical session, held in the Lincoln Room of the Shirley-Savoy Hotel, was called to order at 9:30 a.m. on Wednesday, July 1st by President Taylor. E. R. Queer served as Chairman and three technical papers were presented by the authors and one by title.

President Taylor called the fourth session to order on Wednesday, July 1st at 1:30 p.m. and Chairman L. N. Hunter called upon the authors to present their papers. In addition, one paper was presented by title.

At the conclusion of the session, President Taylor resumed the chair and called upon W. J. Collins, Jr., to present the report of the Committee on Resolutions which follows:

WHEREAS, the A.S.H.V.E. Semi-Annual Meeting 1953 in the mile high City of Denver, is coming to a conclusion; and

WHEREAS, the Rocky Mountain Chapter, through its excellent Committee on Arrangements has made this an outstanding, not to say elevating event; and

WHEREAS, the ladies of Host Chapter have so generously given of their time and efforts to please the members, wives and children; and

WHEREAS, President Reg. F. Taylor and the Sessions Chairmen have conducted all meetings in an exemplary manner so that time was given for social enjoyment and renewal of friendships; and

WHEREAS, the Management and Staff of our headquarters hotel, the Shirley-Savoy and cooperating hotels have been so thoughtful of our every need and comfort; and

WHEREAS, the City of Denver has singularly honored our President, Reg. F. Taylor,

THEREFORE, BE IT RESOLVED,

THAT we, the members of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS here assembled in cool (?), colorful, Colorado, give due recognition to these men and women of the Rocky Mountain Chapter, to the various Committee members and to Chairman Fred Janssen and Vice Chairman E. A. Thompson, and

To the City of Denver, the Denver Convention and Visitors Bureau,

To the trade press and local newspapers for their excellent coverage of our meeting,

To the Medicine Pipe Indian Dancers for their splendid entertainment at Elk Falls Dude Ranch,

We extend to them our heartfelt thanks for a wonderful and varied program of events purposely planned for our pleasure, and further,

We express our appreciation to those contributing papers and discussions at our technical sessions,

To Judge Benj. C. Hilliard for his welcoming address,

To Toastmaster Judge Wilson McCarthy, and

To Under Secretary Walter Williams for his consideration in addressing our Banquet,

To the Shirley-Savoy Hotel and to Pres. Reg. F. Taylor, his officers and committees for another successful meeting.

Respectively submitted,
RESOLUTIONS COMMITTEE
M. W. BISHOP, *Chairman*
W. J. COLLINS, JR.
F. H. FAUST

The resolutions were adopted unanimously and the meeting was adjourned at 4:10 p.m.

PROGRAM SEMI-ANNUAL MEETING

The Shirley-Savoy Hotel, Denver, Colorado—June 29-30, July 1, 1953

SATURDAY—June 27

10:00 a.m. Finance Committee Meeting *(Blue Spruce Room)*

2:00 p.m. Council Meeting *(Centennial Room)*

SUNDAY—June 28

10:00 a.m. REGISTRATION, Shirley-Savoy Hotel *(Lincoln Street Annex)*

10:00 a.m. Committee on Research *(Centennial Room)*

2:00 p.m. Sight Seeing tours of City

4:00 p.m. Chapters Conference Committee Meeting *(Colorado Room)*

MONDAY—June 29

9:00 a.m. REGISTRATION, Shirley-Savoy Hotel *(Lincoln Street Annex)*

9:30 a.m. FIRST TECHNICAL SESSION *(Empire Room)*
Call to order by Pres. Reg. F. Taylor
J. W. James, *Chairman*

Performance of a Forced Draft Cooling Tower, by B. H. Spurlock, Jr., presented by the author

Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F, by Tohru Inouye, F. K. Hick, M.D., S. E. Telser, M.D., and R. W. Keeton, M.D., presented by Mr. Inouye

Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems, by W. L. Rogers, presented by the author

12:00 noon WELCOME LUNCHEON *(Lincoln Room)*
Speaker: Hon. Benj. C. Hilliard, Denver

1:30 p.m. GOLF TOURNAMENT—Lakewood Country Club, Research, Eichberg and Paul Bunyan Cups

1:30 p.m. Children's Entertainment
City Park Museum of Natural History and Zoo

2:00 p.m. TAC on Heat Flow Through Glass *(Blue Spruce Room)*

2:30 p.m. Ladies Style Show and Tea, Park Lane Hotel

3:00 p.m. TAC on Weather Design Conditions *(Centennial Room)*

7:00 p.m. GOOD-TIME PARTY, Shirley-Savoy Hotel *(Lincoln Room)*
Gold Rush Days with Miners, Prospectors, Gamblers, Cowboys and Indians—Dress informal

TUESDAY—June 30

9:00 a.m. REGISTRATION, Shirley-Savoy Hotel *(Lincoln Street Annex)*

TUESDAY—June 30 *(continued)*

9:30 a.m. SECOND TECHNICAL SESSION *(Lincoln Room)*
Call to order by Pres. Reg. F. Taylor
W. F. Friend, *Chairman*
Electrical Analogger Application to the Heat Pump Process, by C. F. Kayan, presented by the author
Design and Performance of a Residential Earth Heat Pump, by Merl Baker, presented by Dr. Y. S. Touloukian
Moisture Movement in Soils Due to Temperature Difference, by W. A. Hadley and Raymond Eisenstadt, presented by Dr. Eisenstadt

12:00 noon Nominating Committee Meeting *(Blue Spruce Room)*

12:30 p.m. LADIES THEATRE PARTY
Matinee performance of the Opera Carmen—Central City Opera House

2:00 p.m. MOUNTAIN TOUR AND SUPPER

WEDNESDAY—July 1

9:00 a.m. REGISTRATION, Shirley-Savoy Hotel *(Lincoln Street Annex)*

9:30 a.m. THIRD TECHNICAL SESSION *(Lincoln Room)*
Call to order by Pres. Reg. F. Taylor
Prof. E. R. Queer, *Chairman*
Design Data for Slat-Type Sun Shades for use in Load Estimating, by G. V. Parmelee and D. J. Vild, presented by R. W. Miller
Automatic Permeance Testing by The Permeometer, by F. A. Joy and A. W. Sherdon, presented by Mr. Sherdon
Thermal Performance of Frame Walls, Part II—Air Spaces Blocked at Mid-Height, by G. O. Handegord and N. B. Hutcheon, presented by Dr. Handegord
Development of Thermal Conductivity Probe, by F. C. Hooper and S. C. Chang, presented by title

1:30 p.m. FOURTH TECHNICAL SESSION *(Empire Room)*
Call to order by Pres. Reg. F. Taylor
L. N. Hunter, *Chairman*
Performance of Warm Air Perimeter-Loop and Perimeter-Radial Systems in a Residence, by H. T. Gilkey, R. W. Roose, and M. E. Childs, presented by F. J. Nunlist, Jr.
Heat Exchanges in a Floor Panel Heated Room, by L. F. Schutrum, C. M. Humphreys, and G. V. Parmelee, presented by Mr. Schutrum
Further Studies of Thermal Characteristics of Plaster Panels, by L. F. Schutrum and C. M. Humphreys, presented by C. M. Humphreys
Heat Flow Analysis in Panel Heating or Cooling Sections—Case II: Floor Slab on Earth with Uniformly Spaced Pipes at the Slab-Earth Interface, by H. B. Nottage, C. V. Franks, L. E. Hulbert, and L. F. Schutrum, presented by title

1:30 p.m. Ladies Matinee Party
Elitch Garden Summer Theatre matinee

4:00 p.m. TAC on Hot Water and Steam Heating *(Blue Spruce Room)*

7:00 p.m. SEMI-ANNUAL BANQUET
Shirley-Savoy Hotel *(Lincoln Room)*
Toastmaster: Judge Wilson McCarthy, Pres., Denver and Rio Grande Western Railway.
Speaker: Walter Williams, Under Secretary of Commerce.
Presentation of Memory Books to Lauren E. Seeley and Ernest Szekely by Pres. Reg. F. Taylor.

COMMITTEE ON ARRANGEMENTS

FRED JANSSEN, *General Chairman* E. A. THOMPSON, *General Vice-Chairman*

B. H. SPURLOCK, JR., *Honorary Chairman*

Banquet—W. R. Egan, *Chairman;* J. F. LaTorra, *Vice Chairman;* Mr. & Mrs. W. J. Barsch, Mr. & Mrs. M. R. Hogue, D. C. Jardine, F. P. Loveland, Mr. & Mrs. J. E. McNevin.

Entertainment—J. R. Wilson, *Chairman;* L. H. Wray, *Vice Chairman;* Mr. & Mrs. D. R. Bachman, Mr. & Mrs. J. F. Cipra, Jr., Mr. R. T. Ekrem, Mr. & Mrs. R. H. Hillbrand, Mr. & Mrs. Ken Murray, Mr. & Mrs. Wallace Olson, Mr. & Mrs. D. D. Pearsall, Mr. & Mrs. K. R. Warshauer, Mr. & Mrs. P. W. Young.

Finance—J. F. Mohan, *Chairman;* L. L. DeLong, *Vice Chairman;* Mr. & Mrs. F. L. Blush, Mr. & Mrs. E. P. Cook, Mr. & Mrs. A. A. D'Amico, Mr. & Mrs. D. M. Frazier, Mr. & Mrs. H. S. Heyse, Mr. & Mrs. C. B. Hickey, Mr. & Mrs. T. P. Jones, Mr. & Mrs. M. D. Klein, Mr. & Mrs. H. C. Kugeler, Mr. & Mrs. J. E. McNevin, Jr., Mr. & Mrs. L. R. O'Rear, Mr. & Mrs. S. W. Rosen, Mr. & Mrs. C. S. Slocum, Mr. & Mrs. A. S. Smith, Mr. & Mrs. J. D. Willoughby, R. L. Vaughan, C. F. Vent.

Ladies—Mr. & Mrs. H. H. Herman, *Chairman;* Mr. & Mrs. P. C. Von Rosenberg, *Co-Chairman;* Mrs. J. W. Braak, Mrs. W. R. Dowdle, Mrs. Leo Krisl, Mrs. R. T. McCalmon, Mrs. R. L. Whittlesey, Mr. & Mrs. G. H. Binning, Mr. & Mrs. H. V. Cook, Mr. & Mrs. A. W. Cooper, Mr. & Mrs. L. V. Davis, Mr. & Mrs. Elmo Hall, Mr. & Mrs. J. J. Johnson, Mr. & Mrs. J. H. McCabe, Mr. & Mrs. E. J. McEahern, Mr. & Mrs. J. F. Mohan, Mr. & Mrs. E. P. Murr, Mr. & Mrs. F. E. Stark, Mr. & Mrs. A. S. Widdowfield, Mr. & Mrs. J. R. Wilson, Mr. & Mrs. H. J. Woehlke, Mr. & Mrs. V. F. Vallero.

Publicity & Attendance Promotion—W. V. Burbank, *Chairman;* J. V. Berger, *Vice Chairman;* C. W. Brown, W. R. Dowdle, E. J. McEahern.

Reception—K. H. Smith, *Chairman;* R. G. Look, *Vice Chairman;* Mr. & Mrs. F. L. Adams, Mr. & Mrs. R. B. Anderson, Mr. & Mrs. G. C. Andresen, Mr. & Mrs. H. G. Angelo, Mr. & Mrs. N. H. Brickham, Mr. & Mrs. L. W. Bryan, Mr. & Mrs. G. E. Eckert, Jr., Mr. & Mrs. E. A. Froese, Mr. & Mrs. S. D. Fox, Jr., Mr. & Mrs. H. D. Gregory, Mr. & Mrs. R. H. Grossman, Mr. & Mrs. G. C. Guenther, Mr. & Mrs. W. B. Heldenbrand, Mr. & Mrs. J. A. McGrew, Mr. & Mrs. J. C. Mechling, Mr. & Mrs. R. W. Petersen, Mr. & Mrs. F. T. Settle, Mr. & Mrs. F. L. Trautman, Mr. & Mrs. K. R. Warshauer, Jr., Mr. & Mrs. D. R. Waterman, M. P. Chapelle, F. R. Pettigrew, J. H. Willis, V. L. Wilkins.

Sessions—F. C. Allen, *Chairman;* H. V. Cook, *Vice Chairman;* F. L. Adams, J. W. Braak, Leo Krisl, R. T. McCalmon, R. L. Whittlesey.

Sports—Elmo Hall, *Chairman;* G. L. Bradbury, *Vice Chairman;* E. L. Kennedy, Jr., H. F. Munn, F. E. Price, E. F. Ross, C. O. Stephan.

Transportation & Special Events—J. H. McCabe, *Chairman;* L. D. Niblack, *Vice Chairman;* Mr. & Mrs. R. T. Beck, Mr. & Mrs. C. L. Boyer, Mr. & Mrs. B. A. Brickham, Mr. & Mrs. L. W. Carter, Mr. & Mrs. J. E. Harrington, Mr. & Mrs. E. L. Heckman, Mr. & Mrs. P. H. Kock, Mr. & Mrs. R. G. Pritchard, Mr. & Mrs. Wm. VanGenderen, Mr. & Mrs. C. B. Ward, J. R. Bernzen, R. W. Booth, W. L. Cook, Jr., H. W. Marshall, C. H. Pollard.

1479

No. 1479

PERFORMANCE OF A FORCED-DRAFT COOLING TOWER

By B. H. SPURLOCK, JR.,* BOULDER, COLO.

THE PERFORMANCE of a cooling tower and especially a forced-draft tower is of interest to many in the field of air conditioning.

Merkel[1] laid the foundations in 1925 for an analytical study of the thermodynamic process involved in the transfer of mass in the tower. Nottage,[2] Mickley,[3] Lichtenstein[4] and others have added to this analysis and have shown that the performance of the tower can be related by the dimensionless quantity

$$\frac{f_m a_m V}{L}$$

The theoretical content of this report has been drawn from the Bibliography. Particular reference is made to the work of Mickley[3] and Lichtenstein,[4] who have made a rather complete analysis of the cooling tower. The reader is recommended to extend his background by use of these and other references.

The experimental research included here covers the effect of spacing of the slat-type packing and the angle of the slats upon the above dimensionless quantity.

ASSUMPTIONS

The following assumptions are considered as a basis of derivation of the equations involved in the cooling tower performance: (1) Tower is forced-draft, adiabatic, counter-flow, and constant cross-section; (2) Enthalpy of gas-vapor mixture may be represented by Equation 1; (3) Evaporation of water is negligible; (4) Variation in humid heat may be neglected; (5) Interaction of simultaneous heat and mass transfer in film may be neglected; (6) Lewis relation is valid and $f_c = C_{ps} f_m$; (7) Area of heat transfer is equal to the area for mass

* Associate Professor, Department of Mechanical Engineering, University of Colorado. Member of A.S.H.V.E.

[1] Exponent numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

transfer, *i.e.*, $a_c = a_m$; (8) Enthalpy of liquid is negligible compared to that of vapor.

BASIC EQUATIONS

It is assumed that the enthalpy of the air-vapor mixture can be represented by the expression

$$h_g = C_{ps}(t_g - t_o) + \lambda_o H_g \quad \ldots\ldots\ldots\ldots (1)$$

where

$C_{ps} = 0.24 + 0.45\, H_g$ = specific heat of humid air.

An enthalpy or heat balance for a differential volume of the tower is given by

$$G\, d[h_g] = d[L\, C_{pL}(t_L - t_o)]$$

which reduces to

$$G\, dh_g = L\, C_{pL} dt_L \quad \ldots\ldots\ldots\ldots\ldots (2)$$

provided the evaporation (small) is omitted. Then

$$\frac{dh_g}{dt_L} = \frac{LC_{pL}}{G} \quad \ldots\ldots\ldots\ldots\ldots (3)$$

The rate of heat transfer can be based on (*a*) (*b*) or (*c*):

(*a*). Heat transfer through the liquid film

$$dq = f_{cL}\, a_c\, (t_L - t_i)\, dV \quad \ldots\ldots\ldots\ldots (4)$$

(*b*). Heat transfer through the air film

$$dq = f_{cg}\, a_c\, (t_i - t_g)\, dV \quad \ldots\ldots\ldots\ldots (5)$$

(*c*). Heat transfer based on the overall coefficient

$$dq = f_c\, a_c\, (t_L - t_g)\, dV \quad \ldots\ldots\ldots\ldots (6)$$

Similar equations for mass transfer may be based on (*a*) (*b*) or (*c*):

(*a*). Liquid film transfer

$$G\, dH_g = f_{mL}\, a_m\, (H_L - H_i)\, dV \quad \ldots\ldots\ldots\ldots (7)$$

(*b*). Gaseous film transfer

$$G\, dH_g = f_{mg}\, a_m\, (H_i - H_g)\, dV \quad \ldots\ldots\ldots\ldots (8)$$

(*c*). Overall transfer coefficient

$$G\, dH_g = f_m\, a_m\, (H_L - H_g)\, dV \quad \ldots\ldots\ldots\ldots (9)$$

Equation 3 may be integrated directly to give

$$h_{g_2} - h_{g_1} = \frac{L\, C_{pL}\, (t_{L_2} - t_{L_1})}{G} \quad \ldots\ldots\ldots\ldots (10)$$

which is termed the *operating line*. The heat transfer across the gaseous film,

$$dq = f_{cg}\, a_c\, (t_i - t_g)\, dV$$

is equal to the rise in enthalpy of the mixture $G dh_g$, or

$$G\, dh_g = f_{cg}\, a_c\, (t_i - t_g)\, dV \quad \ldots\ldots\ldots\ldots (11)$$

NOMENCLATURE

a_c = interfacial area for heat transfer, square feet per cubic foot of tower volume.
a_m = interfacial area for mass transfer, square feet per cubic foot of tower volume.
C_{pL} = specific heat of water, Btu per (pound) (Fahrenheit degree).
C_{ps} = specific heat of humid air, Btu per (pound) (Fahrenheit degree).
f_c = overall thermal conductance, Btu per (square foot) (hour) (Fahrenheit degree).
f_{cg} = thermal conductance of gaseous film, Btu per (square foot) (hour) (Fahrenheit degree).
f_{cL} = thermal conductance of liquid film, Btu per (square foot) (hour) (Fahrenheit degree).
f_m = overall mass conductance, pounds per (hour) (square foot) (unit humidity gradient).
f_{mg} = mass conductance of gaseous film, pounds per (hour) (square foot) (unit humidity gradient).
f_{mL} = mass conductance of liquid film, pounds per (hour) (square foot) (unit humidity gradient).
G = rate of air flow, pounds per (hour) (square foot tower area).
H_g = specific humidity of humid air, pounds water per pound dry air.
H_i = specific humidity of air at interface temperature t_i, pounds water per pound dry air.
H_L = specific humidity of air at liquid temperature t_L, pounds water per pound dry air.
h_g = enthalpy of humid air at t_g, Btu per pound dry air.
h_i = enthalpy of humid air at t_i, Btu per pound dry air.
h_L = enthalpy of humid air at t_L, Btu per pound dry air.
K_m = overall performance factor $\frac{f_m a_m V}{L}$.
L = rate of water flow, pounds per (hour) (square foot of tower area).
q = rate of heat transfer, Btu per hour. (Square foot of tower area).
t = temperature (Fahrenheit).
t_g = temperature of air stream, (Fahrenheit).
t_i = temperature of humid air at interface, (Fahrenheit).
t_L = temperature of water, (Fahrenheit).
t_o = base temperature of enthalpy, (Fahrenheit).
V = volume of tower, cubic feet per (square foot of tower area).
λo = latent heat of vaporization, Btu per pound.

Assuming the validity of the Lewis relation

$$\frac{f_{cg}}{f_{mg}} = C_{ps} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (12)$$

Equation 11 becomes

$$G\,dh_g = f_{mg}\,C_{ps}\,a_c\,(t_i - t_g)\,dV \quad \ldots\ldots\ldots\ldots \quad (13)$$

or, if $a_m = a_c$, then

$$G\,dh_g = f_{mg}\,a_m\,(h_i - h_g)\,dV \quad \ldots\ldots\ldots\ldots \quad (14)$$

and

$$\frac{G\,dh_g}{(h_i - h_g)} = f_{mg}\,a_m\,dV \quad \ldots\ldots\ldots\ldots \quad (15)$$

If it had been assumed that the Lewis relation is given by

$$\frac{t_c}{f_m} = C_{ps}$$

then

$$G\,dh_g = f_m\,C_{ps}\,a_m\,(t_L - t_g)\,dV$$

or

$$\frac{dh_g}{(h_L - h_g)} = \frac{f_m\,a_m dV}{G} \quad \ldots\ldots\ldots\ldots \quad (16)$$

From Equation 2

$$G\,dh_g = LC_{pL}dt_L$$

and Equation 6

$$G\,dh_g = LC_{pL}dt_L = f_e\,a_e\,(t_L - t_g)\,dV$$

and noting that

$$\frac{f_e}{f_m} = C_{ps} \text{ and } C_{pL} = 1$$

$$L \times 1 \times dt_L = f_m\,C_{ps}\,a_m\,(t_L - t_g)\,dV$$

or

$$\frac{dt_L}{(h_L - h_g)} = \frac{f_m\,a_m dV}{L} \quad \ldots\ldots\ldots\ldots (17)$$

From Equation 4

$$G\,dh_g = f_{eL}\,a_e\,(t_L - t_i)\,dV$$

and Equation 14

$$G\,dh_g = f_{mg}\,a_m\,(h_i - h_g)\,dV$$

it follows that

$$\frac{(h_i - h_g)}{(t_L - t_i)} = \frac{f_{eL}}{f_m} \quad \ldots\ldots\ldots\ldots (18)$$

This expression is termed the *tie line* by Mickley.
Also dividing Equation 14 by Equation 11

$$\frac{dh_g}{dt_g} = \frac{(h_i - h_g)}{(t_i - t_g)} \quad \ldots\ldots\ldots\ldots (19)$$

Considering Equation 15, the right side is readily integrated and

$$\int_{h_{g_1}}^{h_{g_2}} \frac{dh_g}{(h_i - h_g)} = \frac{f_{mg}\,a_m\,V}{G} \quad \ldots\ldots\ldots\ldots (20)$$

It is apparent from Equation 20 that the left side is dependent on the thermodynamic properties of the media throughout the tower, while the right side of the equation is a function of the tower design. The performance of the tower is consequently a function of the dimensionless term

$$\frac{f_{mg}\,a_m\,V}{G}$$

Due to the difficulty of expressing the enthalpy of the gas-vapor mixture h_i at the interface, in terms of the enthalpy h_g, it is necessary to employ graphical integration to the left side of Equation 20. Such an integration may be performed by a chart such as shown in Fig. 1, plotting the enthalpy of the gas-vapor mixture h_g as a function of the temperature. The horizontal ordinate may be used to represent the gas stream or the liquid temperature throughout the tower. The curved line, marked *equilibrium line* on the chart, represents the equilibrium conditions for saturated air at the interface temperature and may be

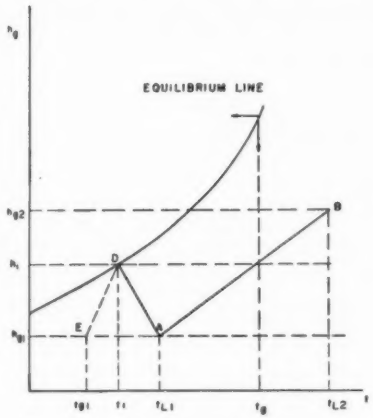


FIG. 1. TEMPERATURE-ENTHALPY DIAGRAM FOR COOLING TOWER PROCESS

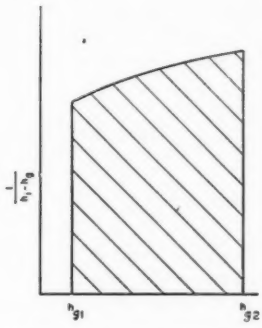


FIG. 2. GRAPHICAL REPRESENTATION OF THE INTEGRAL

$$\int \frac{dh_g}{(h_i - h_g)}$$

calculated directly from tables or taken from the psychrometric chart. The line AB is plotted directly from Equation 10 and represents the end points at the base A and top B of the tower. The slope of the *operating line* AB represents the liquid to gas ratio L/G. Point E represents the condition of the gas entering the tower at the temperature t_{g1} and enthalpy h_{g1}. The point D is located by the slope of the line AD determined by Equation 18. The vertical line from D to the horizontal line AE is the factor $(h_i - h_{g1})$ at the base of the tower, and this quantity evaluated throughout the tower height may be plotted reciprocally on a graph such as shown in Fig. 2. The shaded area gives the value of the integral of the left side of Equation 20.

There are several difficulties encountered in using a graph such as described. For example, the equilibrium line is for values at the interface temperatures and the unit conductances f_{eg} and f_{mg} are the gas side of the interface, and these values are not readily determined. In order to evaluate these terms correctly, it would be necessary to determine the conditions throughout the tower, a time consuming operation at best. Lichtenstein[4] suggests a slightly modified presentation of the tower characteristic. This is obtained by assuming that the *equilibrium line* values are those measured at the temperature t_L rather than the interface temperature t_i. Values of the mass and heat transfer coefficients are the overall values f_m and f_e respectively. Then this gives the expression from Equation 17,

$$\int \frac{dt_L}{(h_L - h_g)} = \frac{f_m a_m V}{L} = K_m$$

A graph similar to Fig. 1 can then be plotted in which the *saturated* or equilibrium values are at the temperature of the liquid t_L and not at the interface

temperature t_1. As shown in Fig. 3, the vertical distance above the operating line to the equilibrium line represents the quantity $(h_L - h_g)$.

On the same chart but with different scale the quantity

$$1/(h_L - h_g)$$

is shown as a function of t_L. The value of the integral is

$$\int_{t_1}^{t_2} \frac{dt_L}{(h_L - h_g)}$$

and is the area under the curve A'B' and above the horizontal axis.

It will be noted in the derivations given that it was assumed that the air was saturated at the interface temperature t_1, while the alternative solution proposed

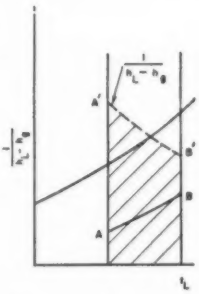


FIG. 3. TEMPERATURE ENTHALPY DIAGRAM FOR OBTAINING OVERALL PERFORMANCE FACTOR K_m

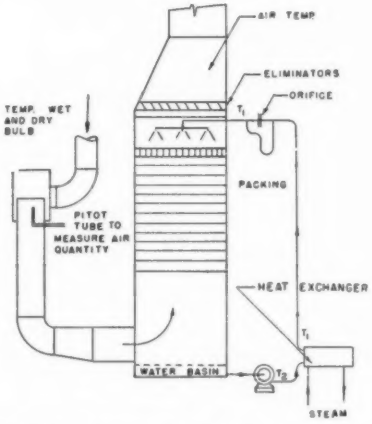


FIG. 4. DIAGRAM OF MODEL COOLING TOWER AND TEST ARRANGEMENT

by Lichtenstein[4] assumes that the saturation temperature of the air corresponds to the temperature of the liquid t_L. Mason[5] indicates that the water film resistance is probably less than 10 percent of the total, and that an error of 20 percent in f_c results in an error of less than 4 percent in the energy exchange. This would seem to justify such an assumption.

Several methods of approach to evaluating this area have been suggested. A common practice is to use the *log mean potential* method, but Lichtenstein[4] points out that the results of this approach are not reliable. Actually little error is involved in assuming a straight line if the range of cooling is not too great, and such a method was adopted in determining the cooling tower performance reported in this paper.

As indicated in the foregoing the quantity $(f_{mg}a_mV)/G$ or $(f_ma_mV)/L$ is a measure of the performance of a cooling tower and is designated K_m. The

accompanying curves indicate the effect of packing spacing and slat angle upon the overall performance factor K_m.

DESCRIPTION OF TEST APPARATUS

The tower had a working height of 16 ft and measured 4 ft x 4 ft on the inside. It was constructed of 1-in. cypress and rested in a galvanized sump-tank on a base 1 ft in height (see Fig. 4). Make-up water was provided through a float valve. Water was taken from the sump by a centrifugal pump and circulated through a closed heat exchanger, through a control valve to the top of the tower, where it was metered by means of an orifice to a header, and then sprayed into the air stream at the top of the tower by means of four conical nozzles.

The tower itself consisted of three sections. The bottom section, resting in the sump-tank, was 6 ft in height with air entering horizontally through one of the sides, where it was turned upward by experimentally located guide vanes. The middle section was 7 ft high, one side of which was a door for entry and on the adjacent side were located brackets to support the packing. The top section of the tower housed the drift eliminators, spray nozzles, and an *egg crate* straightener, located in that order from the top down. The straightener was provided to promote uniform distribution and minimize the water flowing down the side walls.

Air was circulated through the system, by means of a centrifugal fan, and was drawn in from the roof of the laboratory, was conveyed through a 2-ft diameter duct and was introduced horizontally into the tower at the base. The air left the top of the tower and was exhausted to the outside of the laboratory at some distance east of the entrance section.

The packing consisted of a number of trays or rows that were arranged on brackets provided within the center section of the tower. Each tray contained 38 redwood slats, $2\frac{3}{4}$ x $\frac{3}{8}$ in. and slightly less than 4 ft in length. The angle of the slat could be adjusted as desired. The position of each tray was rotated 90 deg about a vertical axis with respect to the tray below.

The essential metering and measuring devices were located as shown in Fig. 4. All temperatures in the tower were measured by proper location of copper-constantan thermocouples. For measuring wet-bulb temperatures, thermocouple junctions were buried in wetted wicks. Wet-bulb and dry-bulb thermometers in the inlet and leaving air were used to check wet- and dry-bulb temperatures. Three wet-bulb and two dry-bulb readings were averaged to obtain the inlet air conditions. Two thermocouple junctions in plastic cones, to keep them dry, and two junctions in wetted wicking located above the eliminators measured the outlet air dry-bulb and wet-bulb temperatures, respectively. A thermocouple sealed in a brass plug and extended into the water stream in the 2-in. pipe near the header section at the top of the tower measured the inlet water temperature. Two thermocouples immersed in the sump-tank at opposite sides of the tower base measured the temperature of the water leaving the tower. The water, controlled by a valve, was metered by a calibrated flat plate square-edged orifice, placed in the line up-stream from the header. The differential pressure was read on a mercury-filled draft gage.

The air flow was determined by a pitot-static tube, located in the center of the duct, after the blower, and connected to a recording draft gage. The average

TABLE 1—TEST DATA 45 DEG SLATS

PACKING FINS INCLINED AT 45 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 78 F

WATER RATE, L = 21,280 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

ROWS OF PACKING	WATER TEMP (T_1)	WATER TEMP (T_2)	AIR TEMP (t_2)	AIR TEMP (t_{w2})	AIR TEMP (t_1)	MASS FLOW RATES (L)	MASS FLOW RATES (G)	HEAT LOST-WATER (Btu/hr)	HEAT GAINED-AIR (Btu/hr)	UNBALANCE (Btu/hr)	% UNBALANCE	EVAPORATION (Lb/hr)	WATER TO AIR RATIO (L/G)	PERFORMANCE $K_m f_m a_m V/L$
0	78.4	64.7	65.0	41.5	60.0	21,280	26,250	281,000	298,500	−17,500	− 6.22	312.7	0.810	0.856
4	77.5	61.8	61.8	39.9	60.7	21,280	24,820	344,000	320,000	23,800	6.93	309.0	0.857	1.086
6	78.0	61.5	63.6	42.0	62.6	21,280	23,850	361,000	317,000	43,500	12.30	306.0	0.892	1.236
8	78.4	60.0	52.0	41.9	63.0	21,280	23,360	399,000	326,200	72,800	18.24	248.5	0.912	1.445
10	78.0	60.0	53.8	42.3	66.4	21,280	23,150	391,000	377,500	13,500	3.45	289.0	0.918	1.657
11	78.4	59.5	54.5	43.0	67.2	21,280	23,150	411,000	412,000	− 1,000	− 0.24	302.0	0.918	1.773
12	78.4	59.3	53.6	42.5	67.6	21,280	23,150	416,000	391,000	25,000	6.01	305.8	0.918	1.910
13	78.4	58.8	55.4	41.5	68.5	21,280	22,250	416,000	406,000	10,000	2.40	328.0	0.955	1.980
14	78.4	59.5	54.0	41.1	68.4	21,280	21,900	412,000	410,000	2,000	4.86	317.5	0.972	1.980
15	77.5	59.0	50.6	36.6	65.4	21,280	21,360	402,000	395,000	7,000	1.74	297.5	0.997	1.536

TABLE 2--TEST DATA 45 DEG SLATS

PACKING FINS INCLINED AT 45 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 96 F

WATER RATE, L = 21,280 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

0	97.6	72.5	52.8	39.0	69.2	21,280	27,150	552,000	559,000	− 7,000	− 1.27	428.0	0.784	0.851
2	96.0	70.8	52.0	38.8	70.2	21,280	25,250	553,000	544,000	9,000	1.63	415.0	0.842	0.937
4	96.4	70.4	58.0	40.4	74.2	21,280	21,650	574,000	609,000	−35,000	− 6.09	489.0	0.863	1.060
6	95.0	67.6	57.2	39.5	74.5	21,280	23,230	601,000	593,000	8,000	1.33	473.0	0.914	1.246
8	96.0	65.0	54.0	38.0	74.1	21,280	23,200	675,500	594,000	81,500	12.08	460.5	0.917	1.415
10	97.6	66.6	50.0	37.5	79.8	21,280	22,570	680,000	734,000	−54,000	− 7.94	545.0	0.943	1.526
11	96.5	65.3	51.5	37.8	78.0	21,280	22,500	672,000	679,000	− 7,000	− 1.42	511.0	0.945	1.528
12	96.6	64.0	51.0	38.5	77.3	21,280	22,480	712,000	661,000	51,000	7.17	496.0	0.946	1.624
13	96.4	63.8	51.0	40.5	79.5	21,280	22,450	712,000	676,000	36,000	5.06	500.0	0.948	1.812
14	96.0	63.4	51.0	40.2	78.4	21,280	22,000	709,000	650,000	59,000	8.42	464.0	0.967	1.782
15	96.0	63.8	52.6	38.5	77.7	21,280	21,250	701,000	628,000	73,000	10.42	460.0	1.001	1.660

TABLE 3—TEST DATA 45 DEG SLATS

PACKING FINS INCLINED AT 45 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 95 F

WATER RATE, L = 18,400 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

ROWS OF PACKING	WATER TEMP (T_1)	WATER TEMP (T_2)	AIR TEMP (t_2)	AIR TEMP (t_{w2})	AIR TEMP (t_1)	MASS FLOW RATES (L)	MASS FLOW RATES (G)	HEAT LOST-WATER (Btu/hr)	HEAT GAINED-AIR (Btu/hr)	UNBALANCE (Btu/hr)	% UNBALANCE	EVAPORATION (Lb/hr)	WATER TO AIR RATIO (L/G)	PERFORMANCE $K_m f_m a_m V/L$
0	94.0	71.2	37.5	33.4	59.2	18,400	26,250	430,000	391,500	38,500	8.83	257.0	0.701	0.702
4	95.0	66.0	46.0	38.2	66.8	18,400	26,200	545,000	493,000	52,000	9.54	341.0	0.702	1.152
8	95.0	61.0	41.6	35.4	70.4	18,400	24,750	635,000	582,000	53,000	8.35	354.0	0.743	1.566
12	94.0	60.4	48.5	36.8	72.4	18,400	24,100	632,000	597,500	34,500	5.38	434.0	0.763	1.725
13	95.0	59.8	47.7	38.8	75.4	18,400	23,620	660,000	633,000	27,000	4.10	452.0	0.778	1.986
14	96.0	61.5	52.7	41.3	76.8	18,400	23,120	649,000	626,000	23,000	3.54	469.0	0.796	1.925
15	94.0	61.0	53.5	40.2	75.2	18,400	22,500	621,000	585,000	36,000	5.79	443.0	0.817	1.895

TABLE 4—TEST DATA 45 DEG SLATS

PACKING FINS INCLINED AT 45 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 95 F

WATER RATE, L = 24,800 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

0	95.0	71.4	45.5	36.5	68.0	24,800	26,850	604,000	557,000	47,000	7.78	388.0	0.923	0.868
4	95.0	68.2	46.2	38.1	72.5	24,800	25,350	679,000	614,000	65,000	9.57	388.0	0.978	1.132
8	95.0	67.8	44.8	39.2	78.4	24,960	24,450	696,000	721,000	−25,000	− 3.59	504.0	1.020	1.364
11	95.0	65.0	44.3	38.4	78.0	24,960	23,680	764,000	708,000	56,000	7.32	490.0	1.055	1.575
12	94.0	60.7	34.3	31.2	77.0	24,800	23,800	840,000	766,000	74,000	8.81	496.0	1.042	1.802
13	93.0	61.0	36.2	32.9	76.5	24,800	23,720	807,000	728,000	79,000	9.78	474.0	1.046	1.800
14	95.0	62.4	36.8	32.5	79.6	24,800	23,500	825,000	818,000	7,000	0.85	547.0	1.055	1.802
15	96.0	61.0	38.5	34.9	81.6	24,800	23,200	885,000	835,000	50,000	5.65	567.0	1.068	2.150

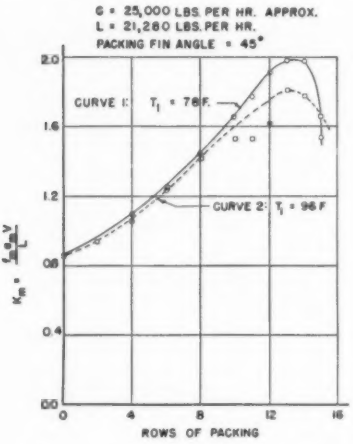


FIG. 5. OVERALL PERFORMANCE FACTOR K_m VS ROWS OF PACKING (PLOT OF THE RESULTS IN TABLES 1 AND 2)

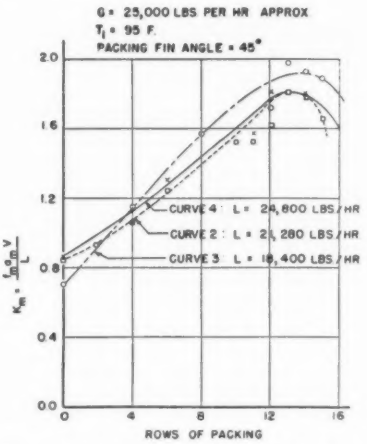

G = 25,000 LBS PER HR APPROX
T1 = 95 F.
PACKING FIN ANGLE = 45°
CURVE 4: L = 28,800 LBS./HR
CURVE 2: L = 21,280 LBS./HR
CURVE 3: L = 18,400 LBS./HR
ROWS OF PACKING

FIG. 6. OVERALL PERFORMANCE FACTOR K_m VS ROWS OF PACKING (PLOT OF THE RESULTS IN TABLES 2, 3, AND 4)

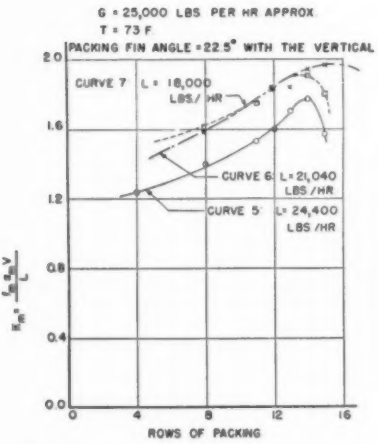

G = 25,000 LBS. PER HR APPROX.
T = 73 F
PACKING FIN ANGLE = 22.5° WITH THE VERTICAL
CURVE 7: L = 18,000 LBS./HR
CURVE 6: L = 21,040 LBS./HR
CURVE 5: L = 24,400 LBS./HR
ROWS OF PACKING

FIG. 7. OVERALL PERFORMANCE FACTOR K_m VS ROWS OF PACKING (PLOT OF THE RESULTS IN TABLES 5, 6, AND 7)

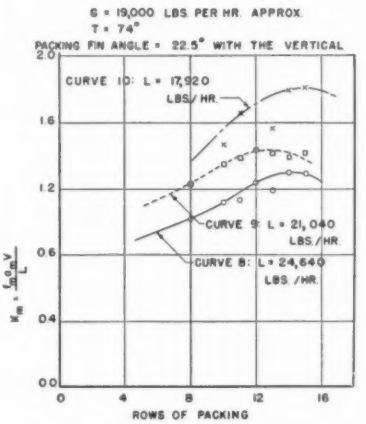

G = 19,000 LBS. PER HR. APPROX.
T = 74°
PACKING FIN ANGLE = 22.5° WITH THE VERTICAL
CURVE 10: L = 17,920 LBS./HR.
CURVE 9: L = 21,040 LBS./HR.
CURVE 8: L = 24,640 LBS./HR.
ROWS OF PACKING

FIG. 8. OVERALL PERFORMANCE FACTOR K_m *vs* ROWS OF PACKING (PLOT OF THE RESULTS IN TABLES 8, 9, AND 10)

velocity was obtained and plotted as a function of the differential pressure measured at the center line and corrected for density. The average velocity was calculated from a 20-position duct-traverse for each point on the curve.

RESULTS

After a number of preliminary tests a general procedure for all runs was established. A range of inlet conditions similar to those that might occur in actual practice was selected. For each set of inlet conditions, a series of runs was made, each following the addition or removal of rows of packing, or a change in the angle of the slat, noting the leaving air and water conditions. When rows of packing were removed the remaining packing was evenly distributed in the tower. Runs were of approximately 30 min in length, to obtain steady state conditions.

As a means of checking the accuracy of the results a check was obtained by calculating a heat balance. Runs with a discrepancy of more than 10 percent, based on the heat given up by the water, were repeated.

Figs. 5 to 8, inclusive, give the effect of the rows of packing and angle of the slat, upon the performance of the tower measured by the quantity, K_m.

Fig. 5 shows the performance for two constant inlet water temperatures 78 F and 96 F, with a slat angle of 45 deg, and a water rate of 21,280 lb per hr and an air rate of approximately 25,000 lb per hr.

Fig. 6 indicates the performance for three different water rates (18,400, 21,280, 24,800 lb per hr) and an air rate of approximately 25,000 lb per hr with an inlet water temperature of 95 F and a slat angle of 45 deg.

Fig. 7 is similar to Fig. 6, but with an inlet water temperature of 73 F and a slat angle of 22½ deg.

Fig. 8 is approximately the same as Fig. 7, but is for an air rate of approximately 19,000 lb per hr.

Fig. 9 gives the effect of changing the slat angle and the amount of packing. The conditions maintained were 75 F inlet water temperature, 21,300 lb per hr water flow rate, and approximately 24,000 lb per hr air flow rate.

Fig. 10 plots the maximum values of the overall performance factor K_m as a function of the water to air ratio L/G.

DISCUSSION OF RESULTS

It should be pointed out that the air-flow rates represented on the curves of Fig. 5 to 10 are approximate values for the range covered. For example, the air-flow rate noted on Fig. 7, Curve 6 is shown as approximately 25,000 lb per hr, whereas actually it varied from 25,400 lb per hr, with 8 rows of packing to 24,300 lb per hr, with 15 rows of packing. This, of course, will affect the results shown in the curves. If Curve 6, Fig. 7, is compared with Curve 9, Fig. 8, both of which are the same except for rate of air flow, it will be noted that for an air flow rate of 24,450 lb per hr (Table 6) K_m has a value of 1.91 and for an air flow rate of 19,680 lb per hr (Table 9) K_m is 1.40, for 14 rows of packing. This indicates an increase of K_m of about 0.1 for each 1000 lb per hr increase in the air flow rate. Hence the curves for constant rate of

TABLE 5—TEST DATA 22.5 DEG SLATS

PACKING FINS INCLINED AT 22.5 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 76 F

WATER RATE, L = 24,400 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

ROWS OF PACKING	WATER TEMP (T_1)	WATER TEMP (T_2)	AIR TEMP (t_2)	AIR TEMP (t_{w2})	AIR TEMP (t_1)	MASS FLOW RATES (L)	MASS FLOW RATES (G)	HEAT LOST-WATER (Btu/hr)	HEAT GAINED-AIR (Btu/hr)	UNBALANCE (Btu/hr)	% UNBALANCE	EVAPORATION (Lb/hr)	WATER TO AIR RATIO (L/G)	PERFORMANCE $K_m f_m a_m V/L$
4	77.8	61.3	48.0	39.8	63.8	24,480	25,900	410,000	379,000	31,000	7.56	284.0	0.945	1.240
8	75.6	59.3	50.0	39.9	62.7	24,480	25,600	406,000	371,000	35,000	8.62	274.0	0.957	1.398
11	75.6	58.2	49.7	39.3	63.6	24,400	25,200	432,000	391,000	41,000	9.50	284.0	0.968	1.534
12	74.0	58.8	57.8	43.2	63.8	24,400	24,600	379,000	339,500	39,500	10.40	285.0	0.992	1.596
13	77.0	59.0	52.9	42.3	66.1	24,400	24,850	444,000	400,000	44,000	9.90	286.0	0.982	1.705
14	77.0	59.5	54.7	44.1	67.0	24,320	24,630	433,000	393,500	39,500	9.12	289.0	0.988	1.769
15	78.0	62.4	62.0	45.6	69.4	24,320	24,050	379,000	416,000	−37,000	− 9.76	354.0	1.010	1.566

TABLE 6—TEST DATA 22.5 DEG SLATS

PACKING FINS INCLINED AT 22.5 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 73 F

WATER RATE, L = 21,040 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

8	73.0	58.5	56.5	45.5	62.6	21,040	25,400	310,000	292,000	18,000	5.80	241.0	0.828	1.622
11	73.5	57.8	55.5	44.3	63.4	21,120	25,150	338,000	321,000	17,000	5.02	258.5	0.839	1.748
12	73.0	57.6	56.1	44.8	63.6	21,040	24,950	330,000	317,000	13,000	3.94	257.0	0.844	1.835
13	74.0	58.5	55.2	43.7	63.6	21,040	24,970	333,000	332,000	1,000	0.30	266.0	0.843	1.680
14	72.0	57.0	60.8	44.1	62.7	21,120	24,450	301,000	300,700	300	0.10	274.5	0.863	1.912
15	72.5	58.2	62.0	45.5	64.1	21,040	24,300	308,000	308,500	− 500	− 0.16	281.0	0.867	1.800

TABLE 7—TEST DATA 22.5 DEG SLATS

PACKING FINS INCLINED AT 22.5 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 73 F

WATER RATE, L = 18,000 LB/HR
AIR RATE, G = 25,000 LB/HR (approx)

8	73.5	56.6	48.0	40.5	60.5	18,000	26,020	310,000	322,500	−12,500	− 4.03	234.5	0.692	1.596
11	72.0	55.0	45.8	40.0	59.8	18,000	25,570	311,000	311,800	− 800	− 0.26	212.0	0.704	1.762
12	72.5	54.5	46.5	40.2	59.5	17,840	25,450	325,000	299,000	26,000	8.00	200.0	0.700	1.838
13	73.5	54.5	45.5	39.0	60.2	17,840	25,350	343,800	329,700	14,100	4.10	228.0	0.704	1.838
14	72.5	53.7	47.1	39.2	59.4	17,920	25,250	341,000	311,000	30,000	8.80	222.0	0.708	1.936
15	73.0	54.0	45.5	39.7	60.2	17,840	25,170	343,500	316,800	27,700	7.77	216.0	0.708	1.968

TABLE 8—TEST DATA 22.5 DEG SLATS

PACKING FINS INCLINED AT 22.5 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 74 F

WATER RATE, L = 24,640 LB/HR
AIR RATE, G = 18,000 LB/HR (approx)

ROWS OF PACKING	WATER TEMP (T_1)	WATER TEMP (T_2)	AIR TEMP (t_2)	AIR TEMP (t_{w2})	AIR TEMP (t_1)	MASS FLOW RATES (L)	MASS FLOW RATES (G)	HEAT LOST-WATER (Btu/hr)	HEAT GAINED-AIR (Btu/hr)	UNBALANCE (Btu/hr)	% UNBALANCE	EVAPORATION (Lb/hr)	WATER TO AIR RATIO (L/G)	PERFORMANCE $K_m f_m a_m V/L$
8	73.5	60.8	45.0	33.0	63.2	24,640	18,860	320,000	343,200	−23,200	− 7.25	245.0	1.307	1.016
10	74.0	60.3	46.5	33.2	64.0	24,640	18,700	347,000	349,700	− 2,700	− 0.78	256.5	1.320	1.118
11	74.4	61.0	48.0	34.0	65.5	24,640	18,620	341,000	366,500	−25,500	− 7.48	276.0	1.325	1.135
12	74.0	60.2	47.8	34.6	65.6	24,640	18,460	349,000	360,000	−11,000	− 3.15	265.0	1.336	1.240
13	74.5	61.0	50.5	35.0	66.1	24,720	18,260	341,000	359,800	−18,800	− 5.51	274.0	1.354	1.188
14	73.5	60.3	48.8	35.9	66.4	24,640	18,080	334,000	353,800	−19,800	− 5.93	261.0	1.364	1.301
15	74.0	60.7	52.0	37.0	66.7	24,640	17,840	337,000	346,000	− 9,000	− 2.67	265.0	1.383	1.296

TABLE 9—TEST DATA 22.5 DEG SLATS

PACKING FINS INCLINED AT 22.5 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 74 F

WATER RATE, L = 21,040 LB/HR
AIR RATE, G = 19,000 LB/HR (approx)

8	73.5	57.6	44.0	31.2	61.1	21,120	20,650	342,000	367,200	−25,200	− 7.37	257.0	1.022	1.235
10	73.5	56.6	42.3	30.4	61.7	21,040	20,230	363,000	376,500	−13,500	− 3.72	257.0	1.040	1.356
11	73.0	55.7	41.8	30.0	60.5	21,040	20,000	370,000	348,000	22,000	5.94	243.0	1.053	1.390
12	74.0	56.0	44.0	30.8	62.6	21,040	19,920	386,000	370,500	15,500	4.01	266.0	1.057	1.438
13	74.0	57.2	42.8	31.5	63.5	20,960	19,880	358,000	378,000	−20,000	− 5.58	266.0	1.052	1.420
14	72.5	56.2	42.8	31.5	61.6	21,040	19,850	348,000	349,500	− 1,500	− 0.43	244.0	1.062	1.400
15	72.5	56.2	42.0	31.8	62.9	20,960	19,680	347,000	364,000	−17,000	− 4.90	248.0	1.063	1.426

TABLE 10—TEST DATA 22.5 DEG SLATS

PACKING FINS INCLINED AT 22.5 DEGREES WITH THE VERTICAL.
INLET WATER TEMPERATURE, T_1 = 73 F

WATER RATE, L = 17,920 LB/HR
AIR RATE, G = 20,000 LB/HR (approx)

10	72.5	55.8	45.5	35.5	49.9	17,920	20,620	304,000	297,200	6,800	2.24	214.0	0.868	1.478
11	72.5	55.0	45.5	35.4	61.2	17,920	20,620	316,000	319,800	− 3,800	− 1.20	228.0	0.868	1.668
12	72.5	55.0	46.8	36.6	61.1	17,840	20,570	317,000	306,500	10,500	3.31	220.0	0.868	1.716
13	72.5	56.0	48.0	38.0	60.6	17,840	20,400	299,000	281,600	17,400	5.82	207.0	0.874	1.572
14	73.0	56.4	51.0	40.3	62.8	17,840	20,250	302,000	290,000	1,200	0.40	220.0	0.880	1.810
15	73.0	56.4	50.6	40.0	63.8	17,920	20,120	302,000	307,800	− 5,800	− 1.92	230.0	0.890	1.830

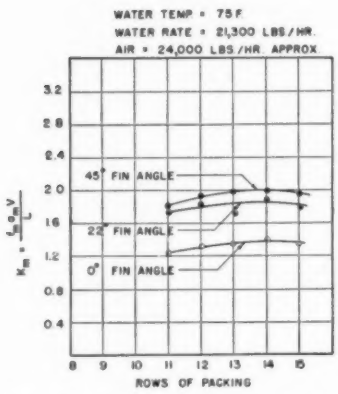


FIG. 9. OVERALL PERFORMANCE FACTOR K_m VS ROWS OF PACKING

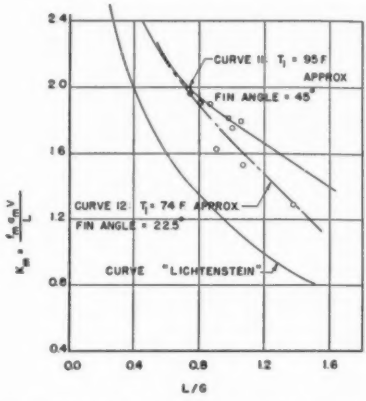


FIG. 10. PLOT OF THE MAXIMUM VALUES OF K_m AS GIVEN BY EACH OF THE CURVES 1 TO 10 VS L/G RATIO FOR THE POINT OF MAXIMUM K_m FOR EACH CURVE

air flow beginning at the initial rate would be expected to fall directly above those shown.

The results in Fig. 10 indicate a higher performance than does Lichtenstein's curve, sketched in for comparison. The discrepancy is in the amount of packing, as Lichtenstein's rows of packing were on 15-in. centers and the present ones correspond to about 6-in. centers. If reference is made again in Fig. 8, to six rows of packing (approximately 15-in. centers) the value of K_m is in agreement with Lichtenstein's results. It would appear that the performance in Lichtenstein's work could be increased by introducing more rows of packing. This would be likely to increase the power requirements considerably. A cost analysis would be required to determine the increase.

It is shown in Fig. 9, that an increase of approximately 30 percent is obtained by slats set at an angle of 22½ deg with the vertical when compared with vertical slats. An additional 12 percent can be obtained by increasing the slat angle to 45 deg. It must be remembered, however, that the power consumption will be increased considerably more for the second 22½ deg than for the first increment.

FURTHER INVESTIGATIONS

This investigation should be extended to cover the effects of different kinds of packing. In addition, a study should be made to determine the part that water running down the walls plays in the overall process. It appears that a higher performance is obtained with a minimum of film surface. Its effect on overall performance should be evaluated.

The assumptions made at the beginning of this paper should be validated not alone for the overall performance, but with regard to their individual effects.

After a complete study is made of the model-type tower, a full scale tower should be tested to ascertain the possible correlation with the prototype.

A study of the power involved will also be necessary in order to establish a rational basis of design.

ACKNOWLEDGMENT

The author wishes to acknowledge the test data shown here as being the cooperative results of graduate students in the Department of Mechanical Engineering at the University of Colorado, and in particular to G. T. Spitzmiller[6] and W. D. Sivers[7].

BIBLIOGRAPHY

1. Verdunstungkuhling, by F. Merkel (V.D.I. Forschungsarbeiten, No. 275, Berlin, 1925).
2. Merkel's Cooling Diagram as a Performance Correlation for Air-Water Cooling Systems, by H. B. Nottage (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, pp. 429-448).
3. Design of Forced Draft Air Conditioning Equipment, by H. S. Mickley (*Chemical Engineering Progress*, December, 1949, pp. 739-745).
4. Performance and Selection of Mechanical Draft Cooling Towers, by Joseph Lichtenstein (*A.S.M.E. Transactions*, October, 1943, pp. 779-788).
5. Performance of a Mechanically Induced Draft, Counterflow Packed Cooling Tower, by A. L. London, W. E. Mason, and L. M. K. Boelter (*A.S.M.E. Transactions*, January, 1940, pp. 41-50).
6. The Vertical Spacing of Slat Type Packing in a Forced Draft Experimental Cooling Tower, by G. T. Spitzmiller (University of Colorado Master's Thesis, 1951, 67 pp.).
7. The Effect on the Performance of the University of Colorado Research Forced Draft Cooling Tower by Varying the Angle of Packing Fins, by W. D. Sivers (University of Colorado Master's Thesis, 1951, 52 pp.).

DISCUSSION

L. T. MART, Kansas City, Mo.: The University of Colorado and Professor Spurlock should be congratulated upon creating interest in a rapidly developing field, and we hope that the studies will be continued and that more colleges will undertake them. The project described in this paper is good for fundamentals and for teaching the subject of cooling towers to students.

Until recently there has been a shortage of graduates who know the basic fundamentals of cooling tower design. This is not surprising when one considers that the cooling tower business has grown from 19 manufacturers, with a gross business volume of approximately three quarters of a million dollars annually in 1933 to a 40 to 50 million dollar volume today. The air conditioning industry alone represents about a third of that. I might say, the tests as conducted were for research and the teaching of fundamentals. The air volumes were very much in accordance with common practice. The loadings were low and rather limited as to gallons per square foot, and encouragement should be given to varying the loadings over much wider ranges. Commercial cooling towers would be of different proportions.

The improvement by inclining the slats is undoubtedly significant but may not be quite correct. I believe that an accurate test will show that the amount of water going to the sidewalls in the three different cases will vary. Such variations might change the values given in the paper for increases in the performance factor.

If the 3 in. slats were cut in half longitudinally and respaced, improved results would be obtained with no additional lumber. Such an arrangement would accomplish

as much as the inclining slats and it is easier to build a tower with flat slats than with the inclined type. I would urge that similar work be done with the crossflow type as well as counterflow towers, as the crossflow towers have become very popular in the smaller sizes and for air conditioning. There are now 12 or 15 manufacturers of crossflow towers, and the performance characteristics are not necessarily alike.

We have seen many large concrete towers but they are not practical in the United States at the present time. The structures are as much as 340 ft high and are seen on large steam turbine works, not on the small tower. A 25,000 kw machine in England would take a tower 250 ft high and 240 ft in diameter at the base. It could have about 8,000 tons of concrete and 1200 tons of reinforcing steel in the shell and the internal part of the tower would have about one third as much. In America, the cost would be very high and could not possibly be justified except in a case where wood could not stand up or where a tower was depreciated over a longer period of years than is common in American practice. They are a fine product, but suitable lumber is scarce in England and on the Continent while steel and concrete are abundant. Thus the use of these latter materials for tower construction is fully justified.

There are many cooling towers in Europe of concrete with no lumber in them and very little steel other than the reinforcing in the concrete. I have seen transite pipe for distribution; also prestressed and precast concrete framing, deck support and splash bars. The small amounts of wood used in Europe are primarily Russian redwood from the northern European or Scandinavian countries.

We are working very closely with the largest manufacturer of concrete towers. It is expected that soon towers using mechanical draft with greatly modified or reduced concrete shells will be available in this country. The use of mechanical drafts will mean lower costs and a notable increase in performance for structures of a given size. Extreme heights of 250 and 300 ft are needed only to create a draft and in America this can be done more economically with fans. The concrete cylindrical shell is thus practical for large industrial forced draft towers in heights of 90 to 100 ft.

In the air conditioning field there have been many special towers built with masonry casings of various materials. Many of the prominent manufacturers will cooperate with the prospective buyer if he wishes a masonry tower which will harmonize with the building. If it is also desirable to hide the mechanical equipment, it can be done.

In the air conditioning field the many types of towers which must be available pose another problem. Whether low or high towers in wood, transite, or steel are considered; each can be either forced draft or atmospheric towers. The possible combinations are so many and varied that engineering overhead is unusually high.

Noise is frequently a serious problem and should be studied. There is but very limited technical information on these subjects to guide manufacturers.

The *Cooling Tower Institute* was organized less than three years ago, and one of the major problems recognized was that of recirculation on the large industrial towers. There is a large tower in the Houston area where very accurate tests will be conducted on recirculation, to determine the extent and the loss due to it, and we hope to obtain practical data from the tests on this tower. Similar tests on recirculation losses are also contemplated for other towers.

Most large towers have air entering from both sides; with only a slight breeze the hot humid outgoing air is usually forced toward one side of the tower, and a portion of it re-enters the tower thus reducing the cooling effect. The cooling tower manufacturers through the *C.T.I.* have demonstrated their concern for this problem by budgeting $37,000 for operations this year.

Another major problem on which considerable money will be spent by the Institute is research on causes of wood decay or rot or fungus attack.

Plastics were mentioned, and have been tried in many ways, but study to date indicates that they are rather expensive. As yet they have not proved practicable primarily because of the difference in cost compared to towers of conventional

materials. Cooling towers for air conditioning usually operate only about 1,000 to 1,500 hr per season and the investment must be economically justified.

W. F. FRIEND, New York, N. Y.: My remarks are based not on an analysis of the paper, but on the general application of the principles of cooling water with air.

Heat dissipation by this method has become the basis of a major industry, to which an important contribution has been made by the many electric generating stations built on sites where there is insufficient cooling water from a stream or from a lake. My own company has had the responsibility for the design and the construction of plants which aggregate several million kilowatts of capacity over the last 20 years. These plants have utilized cooling towers.

In these plants one of the major problems, not covered by this paper, is that of preventing recirculation of the air discharged from the tower back into the intake. At the present time there are several research projects on that subject one of which was mentioned by Mr. Mart in his discussion.

We also have some recirculating problems on air-conditioning towers, where they are located in downtown areas or have been placed within small courts around hotels, and in similar confined locations in industrial plants. One of the major testing projects at the present moment is to get further clues on how air circulates in the vicinity of cooling towers—both in the open and in congested areas, under varying atmospheric conditions.

Inability to predict or to control the movement of air in open spaces was one of the reasons for abandoning the general use of spray ponds, a decade or two ago. The same kind of problems are being encountered, but less seriously, with cooling towers today.

H. E. ZEIL, Detroit, Mich.: There is little that can be added to the information presented by Professor Spurlock, but there are certain conditions affecting the overall size and shape of the tower which are of interest to the architect or engineer.

A study of cooling towers should include methods for establishing some relationship between tower height and tower area. Conditions may exist where it is desirable to reduce the height of the tower several feet and thus the tower base or quantities of air handled must be increased to obtain the desired results.

The use of plastics was mentioned and in this connection fire hazards must be considered.

It would be well if the Society would undertake a study of spray ponds. Very little information is available pertaining to this subject; yet in outlying plants where large water reservoirs are required for fire protection sprays may be used where cooling water is required for plant process or condensers.

W. A. DANIELSON, Raleigh, Tenn.: In Europe and the desert regions the hourglass shape of cooling tower using natural draft is extensively employed. Those dimensions run into over 100 ft by 100 ft. I think one was constructed at the Mayo Clinic some two years ago, and I am wondering if there is any information here on the Mayo experience. I did not personally secure much engineering data on those large towers, but I think they have a great future as indicated by the way they are used in Europe.

G. W. HUMMEL, Phoenix, Ariz.: I would like to ask the author to discuss the gaseous film factor in more detail.

AUTHOR'S CLOSURE: The author is, indeed, appreciative of the comments made by the discussors. It is gratifying to see such interest in a subject that seems to have been neglected for the last few years.

There appear to be two remarks that are pertinent to the discussion. First, it is evident that we are concerned with a most complex problem in any discussion of the cooling tower process. In addition to the thermodynamics involved tower geometry must be considered. When one begins to analyze the effects of the sidewalls, slat

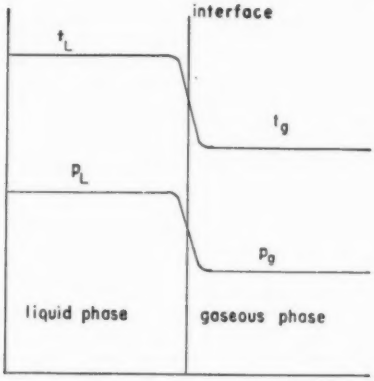


FIG. A

size, slat angle and spacing and the splashing on the slats, the phenomena is beyond comprehension.

Secondly, in studying the cooling tower process, the primary objective of the research engineer is to obtain performance data for the tower that will give the design engineer a clear understanding of what is occurring and enable him to design a tower on a rational basis.

Mr. Mart's ideas are welcomed and constructive. The author would like to make an exhaustive study of the various types of towers but financial considerations limit what can be undertaken. The drift from the tower was of the order of 3 percent or less.

Mr. Hummel's question in regard to the gaseous film factor is explained by Fig. A. In the heat transfer problem of a solid or a liquid to a gas (vapor), there must be a temperature gradient on both sides of the interface, or there would be no heat exchange between the liquid and the vapor at the interface. Assuming that the liquid is in equilibrium or nearly so, there will then be a pressure gradient on the liquid side of the interface. It is this difference in the pressure that gives rise to the mass transfer coefficient on the liquid side. On the vapor side the pressure drops to the partial pressure of the water vapor in the atmosphere and this constitutes the pressure gradient for the mass transfer in the gaseous (vapor) film. The overall pressure drop available for mass transfer is represented by the pressure in the liquid, some distance from the surface, less the pressure in the atmosphere at a distance from the interface. The mass transfer can be expressed in terms of this pressure difference.

Mickley* assumed that the interface was at the saturation temperature of the droplet. This I cannot believe as it would eliminate the possibility of a gradient in the liquid phase.

---

* See Reference 3 of the paper.

1480

No. 1480

EFFECT OF RELATIVE HUMIDITY ON HEAT LOSS OF MEN EXPOSED TO ENVIRONMENTS OF 80, 76, AND 72 F

By TOHRU INOUYE*, F. K. HICK, M.D.**, S. E. TELSER, M.D.† AND R. W. KEETON, M.D.‡, CHICAGO, ILL.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and by the United States Public Health Service in cooperation with the University of Illinois, College of Medicine, Chicago, Ill.

THE PURPOSE of this investigation has been to re-examine the effects of environments with widely different relative humidities on the partition of heat loss of uniformly and lightly clad men and on their subjective sensations of thermal comfort. The environments were selected to slowly cool the skin without inducing visible sweat or apparent shivering during a period of three hours. From a review of results obtained by previous investigators, it is evident that the results were not consistent with those which are obtained by summation of the effect of relative humidity on the nude and of clothing on the nude. A discussion of the tests of previous investigators follows.

The influence of relative humidity on the partition of heat loss of subjects exposed to comfortable environments has been of interest to many observers concerned with heat balance and with thermal comfort. Benedict and Root[1], in an effort to devise a simple method for predicting total heat production, found that heat loss by evaporation of *insensible* sweat varied directly with heat production. This was presented in a graph, an analysis of which showed that for every 2¼ Btu per hr of heat produced in excess of 78 Btu per hr there was 1 Btu per hr of heat lost by evaporation. The ideal conditions recommended for this production of heat were to stabilize the patient just below the sweating

* Research Assistant, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

** Professor of Medicine, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

† Formerly Instructor of Medicine, Department of Medicine, University of Illinois.

‡ Professor of Medicine, Emeritus, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

level, by exposure to an environment maintained at 73.4 F or by adjusting the amount of clothing or bed covering. Wiley and Newburgh[2] demonstrated that relative humidity of the environment influenced the rate of heat loss by evaporation in the nude man. An environment maintained at 82.4 F with a 20 percent RH (relative humidity) in contrast to one with an 80 percent RH induced significantly greater heat loss by evaporation, although the heat production and

TABLE 1—THE PHYSICAL MEASUREMENTS OF SUBJECTS

SUBJECT	AGE YEARS	HEIGHT INCHES	WEIGHT POUNDS	SURFACE AREA SQ FT
Men (fasting)				
V.V.	23	67	155	19.5
L.T.	17	70	137	19.0
A.K.	20	70	125	19.0
A.R.	20	69	197	21.7
B.K.	23	68	141	19.4
Men (non-fasting)				
M.B.	22	67	146	19.1
W.F.	22	68	148	19.4
W.G.	26	72	139	19.6
J.G.	24	71	135	19.2
N.G.	21	65	125	17.4
N.L.	26	66	162	19.8
A.M.	20	66	133	18.2
L.P.	24	72	199	22.8
C.R.	22	73	195	22.8
N.S.	19	69	188	21.6
Women (non-fasting)				
C.C.	19	63	124	16.9
J.D.	24	64	112	15.8
C.H.	21	63	107	16.0
G.K.	24	66	116	16.9
M.L.	19	64	135	17.9
L.M.	20	66	150	19.2
D.M.	21	66	140	18.4
D.N.	20	62	105	15.5
AVERAGE				
Men (fasting)	20.6	68.8	151.0	19.7
Men (non-fasting)	23.6	68.9	157.0	20.0
Women (non-fasting)	21.0	64.3	123.6	17.1

skin temperature remained the same. This effect of relative humidity on the water losses was also shown with small areas of the body by Vasti[3], who used intact skin, and by Pinson[4], who used skin having the sweat glands inactivated. The results on the nude, however, were not substantiated by Winslow and others[5] with their studies by partitional calorimetry. The discrepancies in the results are perhaps explained by differences in procedure. Wiley and Newburgh[2] observed their subject for 150 min thus allowing more time than the other investigators for stabilization and adjustment to the environment.

It was apparent to Freeman and Lengyel[6] that the effects of relative humidity on evaporation demonstrated by Wiley and Newburgh must be reflected in

skin temperatures if heat balance was to be maintained. After their nude fasting men remained for about 30 min in a comfortable environment maintained at 75.2 F with a 20 percent RH, the relative humidity was increased to 90 percent causing the rate of progressive cooling of the skin to be significantly slowed. These results confirmed those of Miura[7] and Phelps and Vold[8] who showed that atmospheres with high relative humidities in contrast to low relative humidities induced higher levels of skin temperature on the face. The effects of relative humidity on skin temperature were, on the other hand, not observed by others[5].

The addition of light clothing to the nude has been shown to raise the levels of skin temperatures without appreciably increasing heat loss by evaporation[2], heat production, or storage[9]. The effects of relative humidity on men dressed in selected street clothes were studied by Houghten and others[10] to determine the environmental factors concerned with heat production and heat loss in order to supply ventilation more conducive to health and comfort. He studied a group of seven healthy young men during a period of four hours in comfortable environments with either 20, 45, 70 or 95 percent RH. These studies showed that heat losses by evaporation, radiation and convection were not affected by the relative humidity. Heat loss by radiation and convection combined was determined by difference between total heat production and heat loss by evaporation. These results on heat losses were duplicated by Wiley and Newburgh[2] who showed, in determining the percentage of heat loss by evaporation to total heat production, that the addition of light clothing abolished the effects of relative humidity observed in their nude subject. Their fasting man was dressed in pajamas and was exposed to either 20 or 60 percent RH in environments maintained as low as 77 F. Similar conclusions were reported by others[11, 12, 13, 14].

Earlier reports from this laboratory[15] indicated that an environment maintained at 76 F with an 80 percent RH produced significantly higher levels of skin temperatures than one at 76 F with a 30 percent RH; without affecting heat loss by evaporation. The subjects were lightly clad and not fasting and were exposed for an hour. A subsequent report[16] using identical experimental conditions failed to confirm these results. It was believed that the discrepancies in the studies were due to the varying degrees of vaso-motor stability and irritability of the subjects as shown by the differences in the levels of rectal temperatures at the time of the exposures. The lack of effect of relative humidity on the skin temperatures of the lightly clothed subjects has also been shown by others[11].

TEST PROCEDURE

This study was carried out in two series of experiments. In the first series the experiments were conducted on five, *fasting*, young healthy men (described in Table 1) during the fall. These men slept over night in a psychrometric room maintained at a uniform temperature of 76 F with either a 30 or 80 percent RH and with an air velocity of less than 20 fpm. The men selected their bed clothing as desired. About 30 min after awakening these men were seated on a balance and weighed at intervals of 30 min for 90 min either in the nude fitted with multiple thermocouples or clothed in union suits rigged with thermocouples[15]. Heat production (basal metabolism) was determined on the men

while in bed and while sitting on the balance at the beginning and end of the observation period. Rectal and skin temperatures were also recorded every 30 min.

In the second series, the experiments were conducted on *non-fasting,* young, healthy males. The men were studied during the winter and spring and the women studied during the summer. Ten men (described in Table 1) were studied in the following manner. The subjects reported to the staff office after breakfast or lunch and sat quietly for 30 min. They then entered the psychrometric room, removed their clothing, dressed in the union suits rigged with thermocouples[15] and sat on the balance for the remainder of the experiment.

TABLE 2—COMPARISON OF OBSERVATIONS ON MEAN SKIN AND RECTAL TEMPERATURES, HEAT LOSS BY EVAPORATION AND HEAT PRODUCTION BETWEEN NUDE AND CLOTHED, FASTING MEN EXPOSED TO ENVIRONMENTS MAINTAINED AT 76 F WITH EITHER A 30 OR 80 PERCENT RH

OBSERVATIONS	PHYSIOLOGICAL ADJUSTMENTS			
	NUDE		CLOTHED	
	30 Percent RH	80 Percent RH	30 Percent RH	80 Percent RH
Mean Skin Temperature F...	90.16	90.42	90.79[a]	91.19
Rectal Temperature F.....	98.02	97.94	97.86	97.74
Heat Loss by Evaporation Btu per (hr) (sq ft).....	4.5	3.2[b]	4.7	3.6[b]
Heat Production Btu per (hr) (sq ft)				
Beginning of test.....	13.4	13.3	13.6	13.3
End of test.....	13.2	13.0	12.8	13.4

[a] Effects of the addition of clothing are statistically significant.
[b] Effects of relative humidity are statistically significant.

The psychrometric room was maintained at either 80, 76 or 72 F with either a 30 or 80 percent RH and with an air velocity of less than 20 fpm. They stayed in each environment for 3 hr, 30 min of which was required to weigh and dress the nude subjects. Eight women (described in Table 1) wearing union suits were studied in the same manner. These subjects were exposed to environments maintained at either 80, 76 or 72 F with a 30 percent RH.

The procedure permitted observations of skin and rectal temperatures, body weights and thermal sensations. Heat loss by evaporation was calculated from changes in body weight assuming minimal errors attributed to respiratory gaseous exchange.

In the first series, the subjects were at basal conditions and their skin temperatures were well adjusted to the environment at the start of the experiment. In the second series, the heat production was approximately 20 to 30 percent above the basal level as the result of their previous activity and the specific dynamic effects of the meal eaten shortly before entrance. On removal of street clothing in no instance was sweating observed on entrance. The subjects

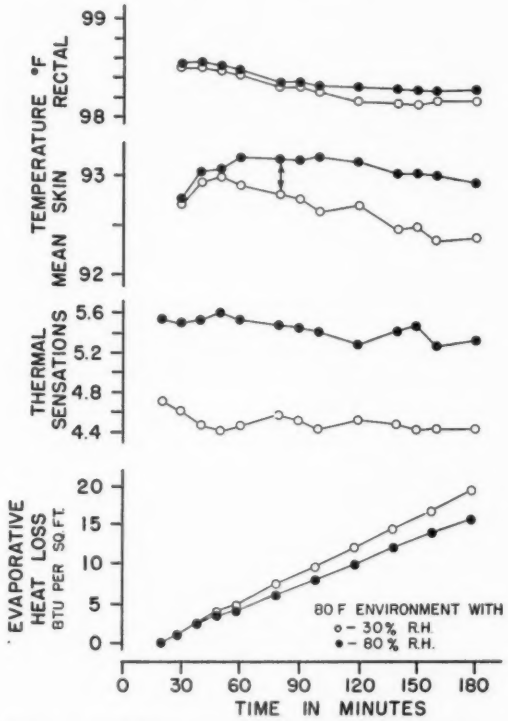


FIG. 1. THE RESPONSES OF MEAN SKIN AND RECTAL TEMPERATURES, THERMAL SENSATIONS AND HEAT LOSS BY EVAPORATION OF MEN DRESSED IN UNION SUITS EXPOSED FOR THREE HOURS TO AN ENVIRONMENT MAINTAINED AT 80 F WITH EITHER A 30 OR 80 PERCENT RH

reported no discomfort. From an inspection of the curves of skin temperature it would appear that the adjustments to change from street clothing to union suits were completed in approximately 15 min after observations were begun.

RESULTS ON MEN

Mean skin temperature: The data on the *fasting* men (first series) obtained about 2 hr after awakening are averaged in Table 2. Heat production is the same in the nude and clothed subjects regardless of the relative humidity of the environment. The heat production taken with the subjects still in bed immediately upon awakening was the same as that taken at the beginning of

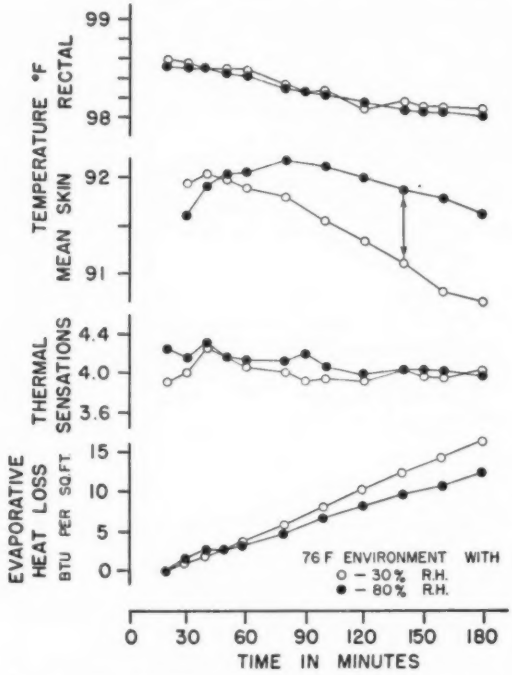


FIG. 2. THE RESPONSES OF MEAN SKIN AND RECTAL TEMPERATURES, THERMAL SENSATIONS AND HEAT LOSS BY EVAPORATION OF MEN DRESSED IN UNION SUITS EXPOSED FOR THREE HOURS TO AN ENVIRONMENT MAINTAINED AT 76 F WITH EITHER A 30 OR 80 PERCENT RH

the observations. It is also the same at the beginning and end of the observation period for each experiment. The mean skin temperatures of similarly clothed subjects were the same in environments with either 30 or 80 percent RH. The mean skin temperatures were higher in the clothed than in the nude only in the environment with the 30 percent RH.

The data obtained on the clothed, *non-fasting* men observed in environments maintained at 80, 76 and 72 F with either a 30 or 80 percent RH are illustrated in Figs. 1, 2 and 3 and are compared in Table 3. The mean skin temperatures were higher in environments maintained at the higher temperature with the same relative humidities. In each environment there were progressive falls in the mean skin temperatures, the greater decreases occurring at the lower

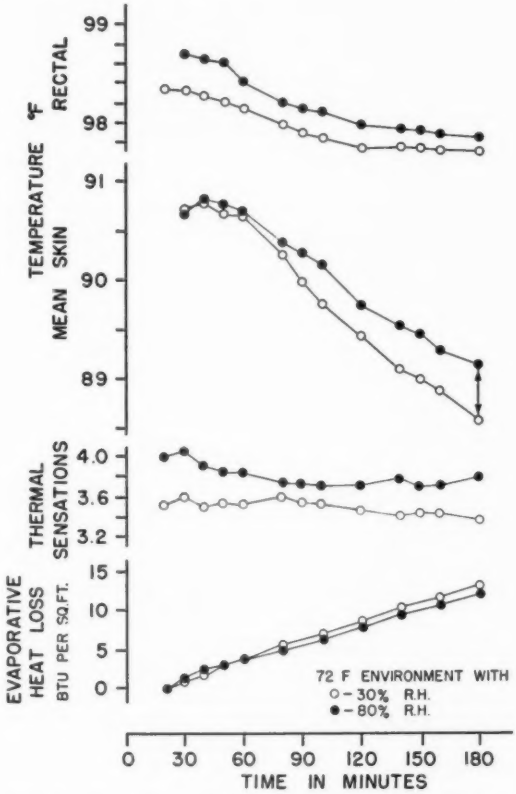


FIG. 3. THE RESPONSES OF MEAN SKIN AND RECTAL TEMPERATURES, THERMAL SENSATIONS AND HEAT LOSS BY EVAPORATION OF MEN DRESSED IN UNION SUITS EXPOSED FOR THREE HOURS TO AN ENVIRONMENT MAINTAINED AT 72 F WITH EITHER A 30 OR 80 PERCENT RH

environmental temperatures. It is apparent in Figs. 1, 2 and 3 that the skin temperatures dropped faster in an environment with a 30 percent than with an 80 percent RH. A statistically significant difference between the mean skin temperatures appeared in 80 min when the environmental temperature was 80 F, Fig. 1. The difference appeared in 140 min when the men were exposed to an environment of 76 F, Fig. 2, and in 180 min to an environment of 72 F, Fig. 3. The time of appearance of a significant difference in the mean skin

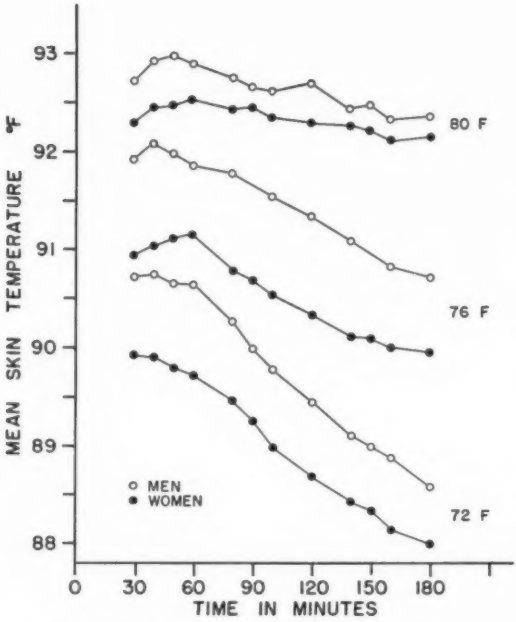


FIG. 4. THE COMPARISON OF MEAN SKIN TEMPERATURES OF MEN AND WOMEN DRESSED IN UNION SUITS EXPOSED FOR THREE HOURS TO ENVIRONMENTS MAINTAINED AT EITHER 80, 76 OR 72 F WITH A 30 PERCENT RH

temperature attributable to relative humidity is indicated in each figure by a double arrow.

Rectal temperature: The rectal temperatures of the men were statistically unaffected by either 30 or 80 percent RH or clothing as shown in Figs. 1, 2 and 3 and Table 2. The greatest decline in rectal temperature was observed in the environment maintained at 72 F, Fig. 3.

Heat loss by evaporation: The rates of heat loss by evaporation of similarly clothed, *fasting* men were greater in environments with a 30 percent rather than in those with an 80 percent RH, Table 2. The rate of heat loss by evaporation was not significantly greater with the addition of clothing when the subjects were exposed to similar environments. This rate might have become significantly greater had more clothed subjects been studied for a longer period of time.

The heat losses by evaporation of clothed, *non-fasting* men are plotted as a function of time in Figs. 1, 2 and 3. At each environmental temperature, the

curves for the low and high relative humidities separated. The greater rates of heat loss by evaporation in an environment with the 30 in contrast to the 80 percent RH were manifested in the second hour when the environmental temperature was at 80 or 76 F and in the third hour when at 72 F. The rates during the third hour of exposure to each environment are compared in Table 4. It was also found that the rate of heat loss by evaporation was greater after three hours of exposure in environments maintained at higher temperatures with the same relative humidities.

Thermal sensation: Immediately upon entrance into the psychrometric room from the staff office, the men still dressed in their street clothes always felt the environment with an 80 percent to be warmer than the one with a 30 percent RH. In an environment maintained at 80 F, the difference in sensations persisted throughout the exposure. After a few minutes in environments maintained at 76 or 72 F these subjects felt thermally comfortable regardless of the relative humidity during the exposure of three hours. Thermal comfort included a range of thermal sensations from *very slightly cool* through *very slightly warm.*

RESULTS ON WOMEN

The data from women in regard to mean skin temperatures and heat loss by evaporation are compared with those of the men in Tables 3 and 4, respectively.

The mean skin temperatures for both sexes are graphed as a function of time (see Fig. 4). It is evident that the mean skin temperatures of women exposed to environments maintained at 76 and 72 F started at lower levels but approached those of the men by the end of the third hour, as shown in Fig. 4 and Table 3. The rates of heat loss by evaporation were less for the women than for the men exposed to similar environments, Table 4. The thermal sensations were the same for the two sexes when exposed to similar environments. The rectal temperatures were also the same except in the environment maintained at 72 F in which the women showed significantly higher values than the men after the first hour of exposure. In this environment the levels were 98.5 and 97.9 F for the women and men, respectively, by the end of the third hour.

DISCUSSION

The test data on the nude fasting men demonstrated that relative humidity affects heat loss by evaporation at a lower environmental temperature (76 F) than those studied by Wiley and Newburgh[2]. The addition of clothing elevated the skin temperatures of these subjects without altering heat production, storage, or heat loss by evaporation. Gagge and others[9] made similar observations. The elevated skin temperatures were necessary to overcome the thermal resistivity of the clothing since the same amount of heat was lost by radiation and convection as in the nude. Clothing reduces the temperature gradients in the skin much in the same manner as elevation of environmental temperatures shown by Bazett and McGlone[17]. The elevated skin temperatures are accompanied by vaso-dilation and an enlargement of the peripheral pool of water available for diffusion. This availability of water for diffusion is confirmed by the greater heat loss by evaporation at higher environmental temperatures even without

free sweating. Burch and Winsor[18] and Pinson[4] have shown that the rate of diffusion of water through the skin was greater with higher tissue temperatures. The escape of water vapor from the surface of the skin into the ambient environment is then complicated by the barrier imposed by clothing. The authors' test results indicate that any resistance of the clothing to the escape of the water vapor is physiologically compensated by increased water vapor pressure at the surface of the skin. This increase in the vapor pressure is just sufficient to maintain the rate of heat loss observed in the nude. In order to demonstrate any effect of relative humidity on these fasting men, who were

TABLE 3—MEAN SKIN TEMPERATURES OF NON-FASTING MEN DRESSED IN UNION SUITS EXPOSED TO ENVIRONMENTS MAINTAINED AT 80, 76 OR 72 F WITH EITHER A 30 OR 80 PERCENT RH. THE VALUES FOR THE MEN AND WOMEN ARE COMPARED WHEN EXPOSED TO ENVIRONMENTS WITH 30 PERCENT RH

ENVIRONMENT			MEAN SKIN TEMPERATURE F				
Temperature F	Percent RH	Effective Temperature	60	90	120	150	180
MEN							
72	30	66.2	90.63[b]	90.06	89.44	88.99	88.58
	80	70.5	90.69	90.25	89.74	89.44	89.14[a]
76	30	68.8	91.86[b]	91.65	91.33[b]	90.94	90.71
	80	73.4	92.06	92.12	91.96	91.79[a]	91.59[a]
80	30	71.8	92.89	92.75	92.69	92.46	92.36
	80	77.5	93.18	93.15[a]	93.11[a]	92.99[a]	92.89[a]
WOMEN							
72	30	66.2	89.72	89.25	88.68	88.32	87.98
76	30	68.8	91.17	90.61	90.32	90.08	89.97
80	30	71.8	92.53	92.43	92.28	92.20	92.13

[a] The mean skin temperature of the men is significantly higher in environments with an 80 rather than one with a 30 percent RH for comparable periods of exposure.
[b] The mean skin temperature of the men is significantly higher than those of the women, also dressed in union suits, for comparable periods of exposure in similar environment.

lightly clothed, the experimental environments must differ from each other by 50 percent in their relative humidities. In the experiments of Wiley and Newburgh[2] the difference in the relative humidities of the test environments was 40 percent.

When the heat production was about 25 to 30 percent above basal, the extra heat was eliminated by evaporation. If greater quantities of heat were produced, other avenues of heat loss would doubtless be requisitioned. The effects of relative humidity on heat loss by evaporation can be more readily demonstrated experimentally when a large quantity of heat is to be lost and when the subject is exposed to higher environmental temperature, providing the latter does not cause active sweating. It was noted in the data that a significant difference in heat loss by evaporation attributable to relative humidity of the environment was shown in the second hour in environments of 80 and 76 F and in the third hour in one of 72 F. With the same heat production in all

the conditions, greater heat loss by evaporation was demonstrated at higher environmental temperatures and their accompanying larger pools of available water in the skin. The reductions in heat loss by evaporation with the elevated relative humidities were accompanied by less cooling of the skin. The effects on the skin temperatures appeared in 80 min in an environment of 80 F, in 140 min in one of 76 F and in 180 min in one of 72 F. The differences in our results with those of Houghten and others[10] are to be explained by the types and amount of clothing. Houghten used a summer clothing assembly consisting of a two-piece cotton underwear, cotton shirt, a two-piece wool suit, socks and shoes; whereas, our subjects wore a single layer, cotton-wool, full

TABLE 4—HEAT LOSS BY EVAPORATION OF NON-FASTING MEN DRESSED IN UNION SUITS AT THE END OF THE THIRD HOUR OF EXPOSURE TO ENVIRONMENTS MAINTAINED AT 80, 76 OR 72 F WITH EITHER A 30 OR 80 PERCENT RH. THE HEAT LOSSES BY EVAPORATION FOR THE WOMEN ARE COMPARED

ENVIRONMENT		HEAT LOSS BY EVAPORATION BTU PER (HR) (SQ FT)	
Temperature F	Percent RH	Men	Women
72	30	4.7[be]	3.7[e]
	80	4.0[ae]	
76	30	6.2[bd]	4.4
	80	4.3[a]	
80	30	6.6[b]	4.8
	80	5.5[a]	

[a] Heat loss by evaporation is significantly greater in environments with a 30 percent than with an 80 percent RH.
[b] Heat loss by evaporation is significantly greater for the men than the women exposed to similar environments.
[c] Heat loss by evaporation is significantly greater in the environments maintained at 80 than at 72 F.
[d] Heat loss by evaporation is significantly greater in the non-fasting than the fasting subjects. The heat loss by evaporation in the fasting subjects who were clothed in union suits was 4.7 Btu per (hr) (sq ft).

length union suit. The lack of effects reported by others[11,12,13,14,16] are explained by differences in either clothing, environmental conditions, physiological state of the subjects, or experimental procedure. Attention is again called to the length of the authors' experimental period which was three hr. This was longer than the time of exposure used by other investigators.

Thermal sensations in the absence of either rapid cooling or rapid warming are essentially a function of the environmental temperature over a range of 76 to 72 F, with either a 30 or 80 percent RH. In the authors' tests men felt equally comfortable in an environment maintained at 76 or 72 F with either a 30 or 80 percent RH. This supplements earlier findings that environments maintained at 76 F with a 30, 60, or 80 percent RH were also thermally comfortable[15]. By changing from a union suit into a summer clothing similar to that used by Houghten the men felt equally comfortable in an environment kept at 76 F with either 30 or 80 percent RH[19]. These results suggest the desirability of determining the extent of variations in relative humidity which may occur in an environment and yet go undetected in thermal sensation and general comfort.

In the present tests data on the women were obtained to establish the patterns of skin cooling and rates of heat loss by evaporation. The women appear to behave differently from the men when they are exposed to environments which allow their body to cool slowly. When contrasted with men in environments of 72 F the heat loss by evaporation was lower, the skin temperatures were the same but the rectal temperatures were higher. It is well known that the heat production is lower in the women than in men[20]. The differences observed could very plausibly be explained by the higher body fat in women [21] which furnishes some insulation.

CONCLUSIONS

Non-fasting men, lightly clad in union suits, showed greater heat loss by evaporation in environments maintained at 80, 76, or 72 F with a 30 percent than with an 80 percent RH. The effects attributed to relative humidity appeared more readily at the higher environmental temperatures.

These men reported that the environments maintained at 76 or 72 F with a 30 percent RH felt thermally the same as one with an 80 percent RH. At the time of entrance into the environment having similar temperatures, they felt warmer in one with an 80 percent than in one with the 30 percent RH. When the limits of themal comfort were set between the sensations of *very slightly warm* and *very slightly cool*, the resting, lightly clad men were thermally comfortable for a period of three hours in environments having temperatures 76 to 72 F and with 30 to 80 percent RH.

The men and women generally showed similar adjustments to environments maintained at 80 or 76 F. When exposed to environments with temperatures of 72 F, the women showed adjustments which were most plausibly explained by their higher body fat.

REFERENCES

1. Insensible Perspiration: Its Relation to Human Physiology and Pathology, by F. G. Benedict and H. F. Root (*Archives of Internal Medicine*, Vol. 38, 1926, pp. 1-35).

2. The Relationship Between the Environment and Basal Insensible Loss of Weight, by F. H. Wiley and L. H. Newburgh (*Journal of Clinical Investigation*, Vol. 10, 1931, pp. 689-701).

3. The Insensible Water Loss Through the Skin, by A. Vasti (*American Journal of Physiology*, Vol. 102, 1932, pp. 60-70).

4. Evaporation From Human Skin With Sweat Glands Inactivated, by E. A. Pinson (*American Journal of Physiology*, Vol. 137, 1942, pp. 492-503).

5. Physiological Reactions of the Human Body to Various Atmospheric Humidities, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 120, 1937, pp. 288-299).

6. The Effects of High Humidity on Skin Temperature at Cool and Warm Conditions, by H. Freeman and B. A. Lengyel (*Journal of Nutrition*, Vol. 17, 1939, pp. 43-52).

7. Effect of Variations in Relative Humidity Upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene,* Vol. 13, 1931, pp. 432-459).

8. Studies in Ventilation. Skin Temperatures as Related to Atmospheric Temperature and Humidity, by E. B. Phelps and A. Vold (*American Journal of Public Health,* Vol. 24, 1934, pp. 959-970).

9. The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures, by A. P. Gagge, C.-E. A. Winslow and L. P. Herrington (*American Journal of Physiology,* Vol. 124, 1938, pp. 30-50).

10. Thermal Changes Between Human Body and Atmospheric Environment, by F. E. Houghten, W. W. Teague, W. E. Miller and W. P. Gant (*American Journal of Physiology,* Vol. 88, 1929, pp. 386-405).

11. The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology,* Vol. 124, 1938, pp. 692-703).

12. Skin Temperature of Children, by F. B. Talbot (*American Journal of Diseases of Children,* Vol. 42, 1931, pp. 965-1052).

13. The Influence of Some External Factors on Insensible Perspiration, by A. van Harreveld, B. W. Grutterink and A. K. Noyons (*Biochemische Zeitschrift,* Vol. 281, 1935, pp. 1-26).

14. Source Factors Determining the Insensible Perspiration of Man, by F. G. Benedict and H. S. H. Wardlow (*Archives of Internal Medicine,* Vol. 49, 1932, pp. 1019-1031).

15. A.S.H.V.E. RESEARCH REPORT No. 1320—Physiological Adjustments of Human Beings to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, S. E. Telser, M.D., R. W. Keeton, M.D., F. K. Hick, M.D., and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, pp. 327-356).

16. A.S.H.V.E. RESEARCH REPORT No. 1380—Physiologic Examination of the Effective Temperature Index, by N. Glickman, T. Inouye, R. W. Keeton and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, pp. 51-74).

17. Temperature Gradients in Tissues of Man, by H. C. Bazett and B. McGlone (*American Journal of Physiology,* Vol. 82, 1927, pp. 415-450).

18. Rate of Insensible Perspiration (diffusion of water) Locally Through Living and Through Dead Human Skin, by G. E. Burch and T. Winsor (*Archives of Internal Medicine,* Vol. 74, 1944, pp. 437-444).

19. A.S.H.V.E. RESEARCH REPORT No. 1447—Physiological Adjustments of Clothed Human Beings to Sudden Changes in Environment—First Hot Moist and Later Comfortable Conditions, by F. K. Hick, M.D., Tohru Inouye, R. W. Keeton, M.D.,

Nathaniel Glickman and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 189).

20. Basal Heat Loss and Production in Women at Temperatures from 23-36° C, by J. D. Hardy and A. T. Milhorat (*Proceedings of the Society of Experimental Biology and Medicine*, Vol. 41, 1939, pp. 94-98).

21. A Method of Breaking Down the Body Weights of Living Persons into Terms of Extracellular Fluid, Cell Mass and Fat, and Some Applications of It to Physiology and Medicine, by R. A. McCance and E. M. Widdowson (*Proceedings of the Royal Society*, Series B, Number 890, 1951, pp. 115-129).

DISCUSSION

L. P. HERRINGTON, New Haven, Conn. (WRITTEN): This is an excellent study of a problem complicated by considerable physiological variability when the subject is nude. When subjects are either clothed or nude, the effects of the hydroscopic properties of skin or clothing, related both to regain properties and to questions of moisture equilibrium present many difficulties.

In the work of the Pierce group, we have recognized that modifications of humidity with constant dry bulb are associated with changes in insensible rate for a transition period, the length of which is a complicated function of the subject's activity and exposure prior to the experiment, the absolute level of temperature, the air velocity, and the weight and fiber character of any clothing worn. After complete adaptation to a comfortable environment, we have found only minimal differences in insensible loss for large changes in humidity at dry bulb in the region of insensible loss.

We regard the question in all such experiments as being that of providing data for an eventual multiple factor prediction of skin temperature and vapor loss rather than one of answering the titular question categorically for the innumerable ambient and subject conditions which are possible, although in the present instance such an interpretation is of practical interest.

The experiments reported here provide valuable data for the quantitative derivation of such an equation from the combined work of many experimenters. This end would be facilitated by publishing a description of the weight and fiber character of the suits worn, their regain behavior when heated internally to 90 F in the test environment, and a chart giving the time course of the subject's weight loss. Black body temperatures and air velocity data should be included if possible. Apart from obvious civil value, military interests in the equational features of this type of adjustment are considerable, and careful studies of this type can be added to current analyses if the accessory data mentioned can be made available.

M. K. FAHNESTOCK, Urbana, Ill.: I would like to speak from the standpoint of the Technical Advisory Committee on Physiological Research of which I am chairman. It would be well to mention that two-thirds of this committee's members are biologists, doctors of medicine and physiologists, and the remainder are engineers. The membership includes some of the best men in this country in the fields of medicine, basic physiology, industrial hygiene and engineering. The committee serves as a clearing house where certain problems of the engineer involving health and comfort

can be brought to the attention of biological people. It is difficult to assemble the members of this committee because they do not attend any common national meeting. I believe that each and every one of us, whether we are in commercial or residential air-conditioning, realize more and more that the indoor conditions which we are creating synthetically have to be, in the long run, conditions which are conducive to health and comfort. It is no longer enough to say that we are comfortable. We must also be sure that people are healthy in these various environments.

This paper is one of a number giving the results of work which the Society started in 1935 and the Society should be commended for realizing the health aspects of air conditioning at that time. Interest in the work was initiated by a small group. One of the difficulties, of course, is to get sufficient money, because physiological work with human subjects is expensive.

The speaker qualified his remarks by pointing out that the results were from tests conducted with sedentary individuals. When one considers the total population of this country and the entire market for air-conditioning it will be realized that many individuals are not sedentary. Therefore, we need to expand the work to include people who are doing work and we need additional financial support for this expansion.

It is rather embarrassing to me as a member of the staff of the University of Illinois to have to relate that most of the work of this TA Committee on Physiological Research has been done at the University of Illinois. I would be in a much more comfortable situation if I could report that work was going on at three or four other recognized medical schools.

One of the Committee's problems is to have the engineers present their problems. This committee is well qualified to consider problems from the biological viewpoint and to render opinions as to whether health is likely to be involved. I appeal to all of you here to give us your problems, and I think that we can give you some of the answers even though we meet only once a year. Your cooperation would help us in outlining needed research programs. The subject of health and comfort is becoming more and more important as air-conditioning is applied to homes where we have old people, young people, ill people, and partially impaired persons, in addition to normal healthy individuals.

The paper just presented and those preceding it included subjective data, given by the subjects regarding their feeling of warmth and coolness, and objective measurements, such as skin temperature, blood pressure, pulse rate, etc. Attempts have been made to correlate the two types of data and thought has been given to the measurement of stress at conditions beyond the comfort range. However, within the range of the conditions studied the experimenters were not successful in establishing the existence of physiological stress by the physical measurements of blood pressure, pulse rates, respiration rates, skin temperature, etc.

Some people believe that environmental stress may possibly be chemical, such as excessive secretions from certain glands. The experience of clinical doctors shows that the general public becomes exhausted during prolonged periods of hot weather. Impaired individuals or others with stress of disease or surgery may die. These facts and indications open up an entire new field of research which is more complicated than the one involving physical physiological measurements.

Again I would like to ask, that as engineers, you bring your questions regarding the possible effects of environmental factors on health to the TAC on Physiological Research. In many instances I believe that the Committee or some of the members can be of help. For example, only today a close personal friend of mine, who has been in the heating industry for years, expressed the opinion that while sleeping he could catch cold on account of the temperature or humidity in an air-conditioned space and that on this account he usually turned the unit off. Those of us who have been studying the effects of temperature and humidity on people are convinced that in a proper air-conditioned space a normal individual will not take cold from staying

in the space or from going in and out of the air conditioned space. However, we do have a feeling that drafts present in many poor air-conditioned installations may cause persons to take cold.

F. H. FAUST, Bloomfield, N. J.: This Society has established an enviable reputation in the field of physiological research and this paper is an important additional contribution from a number of aspects. From the Society's standpoint it further enhances its reputation; from the users' standpoint it has established a little more definitively the combination of temperature and humidity conditions which are desirable for comfort; while from the manufacturer's point of view it may affect the design of equipment, and from the contractor's point of view it may affect the selection and sizing of equipment.

The Joint Committee on Minimum Requirements for Comfort Air Conditioning composed of representatives of A.S.H.V.E. and *A.S.R.E.* established a code for minimum requirements for comfort air conditioning in 1938. Recent studies and developments have made apparent the necessity for revision of this code and a joint committee composed of representatives of A.S.H.V.E., *A.S.R.E., A.S.M.E., A.R.I.,* and the medical profession has been founded for this purpose. It is headed by W. L. Fleisher.

The studies described in this paper are a further step to help in establishing the optimum conditions of temperature and humidity for comfort and health. We are also interested in other aspects such as the effects of radiation, air movement and other things on physiological comfort. There is still much research to be undertaken and reported and we hope that studies of this nature will be continued.

P. J. MARSCHALL, Chicago, Ill.: The authors are to be complimented on the very interesting presentation of this important phase of research having to do with physiological reactions to various conditions of temperature and humidity. The tests reported in the paper cover a period of three hours. There appears to be very little difference in the thermal sensation from start to finish over this period. Can we assume that this would be true for a longer period of exposure, say 8 hrs? This assumption may be questionable in view of the slope of the curve for skin temperature and evaporative heat loss. In the tests reported, the subjects were at rest. I believe we should have further data with subjects at various degrees of activity.

J. DONALD KROEKER, Portland, Ore.: I feel that we need more such information as presented by the authors of this paper. We need to have existing information on comfort requirements reestablished and revised. We are looking for studies such as this, together with reports on studies of persons under normal activity—to guide us in the selection of equipment and in design.

W. A. GRANT, Syracuse, N. Y.: This is another splendid paper by the University of Illinois. While the major emphasis concerns the effect of relative humidity on heat loss, a valuable by-product is the effect on thermal sensation. These results seem to be an additional indication that the A.S.H.V.E. Comfort Chart which engineers use every day contains some serious errors. At 76 F and 72 F with the widely different humidities of 30 and 80 percent, there is an equal sensation of coolness or warmth. It is only when the temperature gets up to 80 F, that the authors obtained a significant difference in thermal sensation at these two humidities, and even then, as I understand it, there was no visible sweating.

In my opinion this indicates that a considerable amount of work should be done on the problem of thermal sensation, to get a more accurate comfort chart. Obviously the effective temperature lines on the present chart cannot be right if the thermal sensation is substantially equal over a large range of temperatures with such widely different humidities. We hope the Society membership will agree that a high priority should be accorded a full-scale research program along these lines.

G. W. HUMMEL, Phoenix, Ariz.: The importance of studies as described in this paper cannot be overemphasized. Particularly important, in my opinion, is research on the physiological effect on individuals upon entering and leaving air conditioned spaces.

We have, as a Society done work in this field, but the studies are not yet complete. The studies should be extended to include sick people, cardiac, obese and other types of individuals under a physician's care.

C. B. GAMBLE, New Orleans, La.: I cannot believe that relative humidity has no effect on an active person. My views are based on observations in many air conditioned buildings. With a relative humidity of 80 percent at 76 F the thermal sensation is *hot*. In our section of the country no one would want these conditions.

I feel that a reader of this paper may get the impression that humidity has very little effect; while that may be true under basal, I don't think it is true under normal conditions.

P. E. MCNALL, JR., Minneapolis, Minn.: The authors are to be complimented on a fine paper. While this paper is concerned with humidity effects, we have been attempting to measure mean radiant temperatures in enclosures. The authors state that radiation losses account for approximately half the total body heat loss under the conditions of the tests. We feel a correlation of mean radiant temperature and air temperature, if possible, is a very important project.

Our preliminary tests have indicated some interesting things. One of these is that scattered solar radiation entering a large picture window on a winter day may more than offset the *cold wall effect* close to the window, causing an apparent mean radiant temperature 1 to 1.5 deg higher than that in the center of the room.

It would be desirable to extend the study reported in this paper to cover people who are infirm, or engaged in physical activities. I think that we are still a long way from understanding comfort for sedentary persons which I feel is the most important case.

P. B. GORDON, New York, N. Y.: This is the nature of a practical observation. We have noted some serious objections in the control and operation of air conditioning systems particularly in office buildings with eight-hour occupancy where the occupants are subjected to a change of 2 or 3 deg in temperature. If the occupants seem to be most comfortable at 75 or 76 F and the temperature should rise to 79 F, it is immediately apparent and usually causes many complaints. Conversely, if the relative humidity due to some operating error or other difficulty should rise to 60 or 70 percent, it is not ordinarily noticed. Evidently in the cases we have observed constant temperature plays a more important role in comfort than constant relative humidity. This substantiates some of the data presented in this paper, although perhaps somewhat different than the experience in New Orleans as described by Mr. Gamble.

AUTHORS' CLOSURE (Tohru Inouye): The adjustments in heat regulation of resting men and women seated for three hours in environments of different relative humidities have been described. During the early period of these exposures the physiological adjustments were influenced by the experimental procedure in which the subjects were required to change from street clothes into the standard test clothing, that is, the union suit. This generally explains the rise in the skin temperature during the first hour when the subjects were exposed to cool environments.

The adjustments are also affected by a number of factors of which physical activity presents a significant problem for the engineers and infirmities in the heat regulation to the clinician. In the case of physical activity the adjustments would depend upon the rate of heat production, from activity and its means of dissipation. From information available in the literature, sweating and its evaporation become the

predominant mechanisms of heat regulation. Environmentally, relative humidity can be expected to take an active role in the dissipation of this heat from the body.

The degree of activity is also a function of the duration of exposure as it contributes to the maintenance of the rate of heat production during the course of exposure. Completely inactive subjects would tend to have the rate of heat production decrease with progress of the exposure in the environmental temperature ranges used in these studies. Practically, however, one rarely can sit quietly for any length of time which, perhaps, accounts for the similarity of heat production at the beginning and end of the three hour tests here reported. This phenomenon could easily be exaggerated by exposures which extend for as long as 8 hours; however, diurnal variations in the metabolic rate can easily become significant factors.

The effect of various environments on the ill is currently under study. It is hoped that patients with defects in heat regulation could afford more information in the inner workings of the heat regulatory mechanisms. This in turn could be used for their management.

The over-all integration of the effect of environment on the human is expressed in the sensation of comfort. This subjection indication is still the best measure of one's response to any environment. Until such time as objective measures can be devised to measure the sensations of comfort, a subjective delineation of the comfort zone should be obtained for men and women according to types and degrees of activity. This would constitute a very valuable and practical extension of our present knowledge.

1481

No. 1481

EXPERIMENTAL APPROACHES TO THE STUDY OF NOISE AND NOISE TRANSMISSION IN PIPING SYSTEMS

By W. L. ROGERS*, EVANSTON, ILL.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Northwestern University, Evanston, Illinois.

THE GENERATION and transmission of noise in water piping and heating systems has always been of vital concern to the heating and plumbing industries. Recognizing the need for more technical information on this subject, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in cooperation with Northwestern University, has sponsored basic and specific research in this field. Experimental equipment was designed and constructed to enable investigation of noise characteristics of straight runs of pipe, pipe fittings, and the insertion of pumps.

Before definite experimental set-up could be established, preliminary study and experimentation were necessary. The first requirement involved control of water pressure, temperature, and velocity in the piping components under test. It was further necessary to recognize that any acoustic disturbances originating in the water, upstream or downstream, from the test components, could be carried into the test region by both liquid and pipe wall vibration. This required the elimination of all pumps for the series involving only noises arising from flow through pipes and fittings. It posed problems of flow control rates and pressures.

Cold water from city supply lines was fed to a 20-ft length of 1-in. pipe. Several feet of soft rubber hose without cord plies were inserted, both before and after the test pipe. It was used to damp out pressure fluctuations in the water coming from upstream. It also prevented supply and discharge pipe wall vibration from reaching the test section. Isolation from building vibration was achieved by suspending the pipe from rubber loops. Under certain conditions audible noises were generated by flow through control valves, elbows, and the orifice used to indicate water flow rate.

Tests were conducted in an air-conditioned room with heavy walls lined with 6-in. cork and faced with cement plaster. Runs were made at night in an

* Assistant Professor of Mechanical Engineering, Northwestern University.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

attempt to achieve a low enough ambient sound level to enable detection of weak flow noises. Noise was measured by means of a microphone placed near the fitting under test. The preliminary tests indicated that certain modifications were desirable. The controls were placed outside the test room; the orifices used as flow-rate indicators were eliminated, and a large tank with controlled air pressure on the water surface inside the tank was adopted as a water supply, thereby making pumps unnecessary. The sound-measurement technique was changed to include a means of measuring the sound-pressure level in the water inside the test pipe and the vibration of the pipe wall.

EXPERIMENTAL SET-UP

Fig. 1 is a schematic diagram of the present test set-up. Water is supplied to the test region from a 4-ft diameter, 8-ft high tank with approximately 750 gal capacity. Desired temperatures are obtained by injecting steam into the water. Desired pressures are produced by admitting compressed air into the tank above the water surface.

The feed line from tank to test region consists of approximately 20 ft of 3-in. pipe, followed by 35 ft of 1½-in. pipe, and finally by 20 ft of steam hose.

Basic test section length is 20 ft with the pipe elastically supported at four points by rubber loops. For testing fittings, the pipe is broken at the middle to allow a 10-ft section both before and after the fitting. The system accommodating test pipe and fitting sizes from ½-in. to 1½-in. can be supported in a live-sound chamber to permit measurement of air-borne sounds. It can be supported outside the chamber where it is more convenient to measure pipe-wall vibration. Special care was used in reducing the outer diameter of the inlet and outlet ends of the test pipe so that the steam hose fits snugly over the pipe ends, without abrupt change in section.

The outlet line from the test region is approximately 50 ft of steam hose, discharging into a 3-in. waste drain line. The water is not recirculated.

OPERATION OF TEST EQUIPMENT

The manner of operation of the test equipment involves a batch process. The tank is first partially filled with water from city supply lines. Then live steam is passed into the water until the desired temperature is reached. When steam is shut off, compressed air is admitted to the tank space above the water until tank pressure is somewhat above desired pressure at the test region. The system is then ready for a run.

Two valves, placed as far as possible from the test region, one after the tank outlet and one at the final discharge to drain, can be used to control pressure and flow rate (velocity) in the test region. In a test run, the two control valves are adjusted to give desired pressure and flow rate. Then, with final drain valve setting locked, manual control of the tank discharge valve is used to maintain desired pressure and flow rate in the test section. The tank valve is opened gradually as tank water level recedes and air above the water expands. Additional air can be added very quickly if air pressure in the tank drops to too low a value to maintain desired pressure in the test region. A run is terminated when the water supply in the tank is exhausted.

The tank is in an enclosure separated from the room enclosing the test section. A gage glass on the tank side allows observation of water level inside

for timing volumetric flow rate. Fig. 2 shows the test section, illustrating the method of hanging the pipe, and some of the instrumentation. Water enters the room from below, rising in the pipe near the middle of the test section. The water is led through steam hose and then enters the test pipe from the left in the picture. At the far side of the test pipe, steam hose carries the water away through the floor, where it discharges to drain in a different room below. Thus both the supply tank and drain region are in rooms separated from the test room.

Runs of about 20 min duration can be made with a velocity up to 10 fps in 1-in. pipe, and proportionate times for other sizes and for other rates. Runs have been made with water temperatures as high as 250 F and with velocities above 10 fps in 1-in. pipe. Pressure in the test region has been maintained

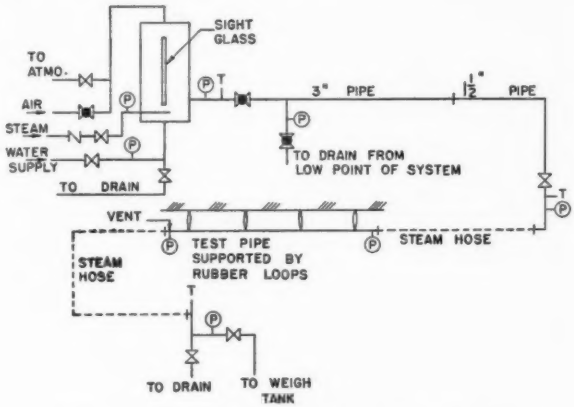


FIG. 1. DIAGRAM OF THE EXPERIMENTAL PIPING ARRANGEMENT

in the 15-30 psig range. Pressure gages and thermometers are used to indicate pressures and temperatures of water in the test region.

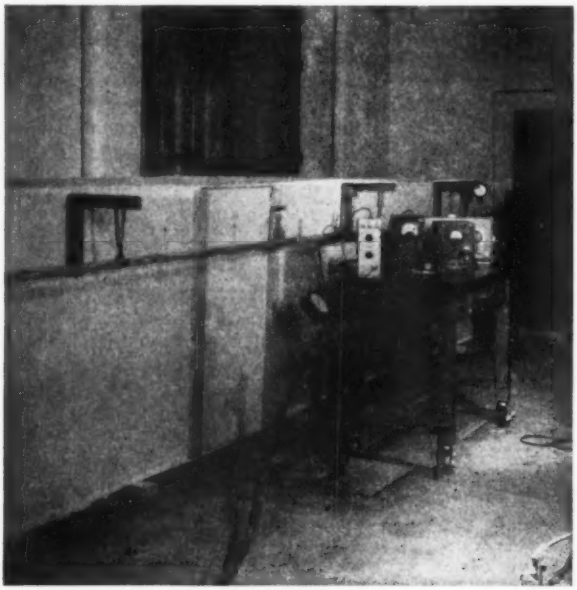
Measuring Water Flow: Several methods of measuring the rate of water flow were considered. It was at first deemed desirable to have an instantaneous reading indicating the flow rate, as was previously mentioned, and calibrated orifices were inserted in the pipe line leading from the test region to drain.

In exploratory runs it was found that under some conditions flow through the orifice plates caused intense noise generation which could be heard clearly (by stethoscope) in the pipe at the test location. Consequently it was decided to eliminate the orifices and measure the flow rate by timing the volume decrease of water in the tank.

Pressure Drop at Discharge Valve: Another problem was created from the sudden pressure drop of hot water through the valve at the final discharge to drain. Noises generated there were transmitted in large measure back to the test region when an insufficient length of rubber hose existed between the test pipe and the drain valve. Considerable experimentation was necessary to deter-

mine what length and size of hose should be used to give effective sound attenuation, without necessitating excessive pressures in the test region during test runs. Another factor involved was the attempt to eliminate the sudden pressure drop at the drain valve, and to substitute the graduated pressure drop which occurs in the discharge rubber hose. More rapid pressure drop gradients seem to generate excessive noise. An effective compromise has been found through adjustment of the length and size of steam hose now being used between the test pipe and the final drain valve.

The problem of keeping the feed water to the test section relatively free of acoustic disturbances has been mentioned briefly. It will be noted that no



(A valve under test is shown at the middle of the pipe)

FIG. 2. THE TEST-SECTION OF PIPE AND SOME OF THE INSTRUMENTATION

pumps are operating during a test run; further, only two valves are used to control pressures and flow rates, and these valves are placed as far from the test location as possible. The lengths of rubber hose used between the valves and the test section contribute to sound attenuation by a pressure-relief and damping effect.

If pressure pulsations (sound waves) reach a location where there is relative freedom of expansion, then a weakened sound wave travelling at reduced velocity leaves the locality. High-frequency pulses are damped out more readily and more rapidly than those of low frequency by the effect obtained in rubber hose. For cold-water runs, soft rubber tubing with no cord insert was found

to be very effective in blocking upstream and downstream sounds from the test pipe. For hot-water runs, however, safety dictated the use of steam hose which has several plies of cord. This type of hose is much stiffer than the soft tubing and is less effective in attenuating sound. To offset this, many more feet of the steam hose are now used than were necessary for the soft tubing.

INSTRUMENTATION FOR SOUND

When sound instrumentation is considered, it must be realized that the vibrations (sounds) produced by the piping components being tested can appear either as fluid-borne or as structural-borne vibration. Part of the sound energy causes the walls of the pipe and fitting to vibrate, and this structural vibration, which is readily transmitted by the structural components, produces air-borne noise. The remainder of the sound energy remains in the liquid and is transmitted in the liquid to other regions where it might eventually become air-borne.

There is relatively little attenuation of sound as it travels in the water inside metal pipes and the sounds may be transmitted long distances without severe loss of energy. Both aspects of the sound transmission are of interest—the immediately air-borne sound from vibrating pipe walls, and the *potential* air-borne and wall-vibrating sound which is carried within as well as along the pipe.

Some of the preliminary measurements of the air-borne sound were made with a microphone placed inside a live sound chamber enclosing the test pipe. In operation, the sound level with no flow was determined, then a 20-ft straight run of pipe was tested and finally various fittings were introduced. A practical difficulty was the attainment of a low enough ambient sound level to permit accurate measurement of weak sounds, as well as other difficulties associated with the frequency characteristics of such a hard-surfaced chamber.

It was decided to try a different approach to the measurement problem. The present system utilizes an accelerometer mounted on the pipe to indicate pipe-wall vibration, and a microphone mounted through the pipe wall in contact with the flowing stream. The reading taken from the accelerometer serves as an indication of air-borne sound (and also structural-borne vibration). The positioning of the vibration pickup (accelerometer) presents a problem, and it is necessary to adopt a composite average of the vibration recorded for various positions of the pickup along the pipe.

The sound level remaining in the water is measured by means of a microphone mounted through the pipe wall into the flowing stream. It must be recognized that the obstruction into the stream may result in sound generation at the microphone. However, preliminary tests have indicated that water flow in a straight pipe run is relatively quiet. The microphone mounting position is fixed in a straight run, therefore, and a reference level is obtained. Noise generated by fittings is then compared with this reference level, and the relative noise-producing characteristics of the fittings can be determined.

The straight pipe run is used as the basic reference level above which both microphone and accelerometer readings produced by fitting noises are measured and compared. The microphone being used is equipped with a DKT (dipotassium tartrate) crystal which exhibits no electrical conductivity as temperature is raised, making the microphone characteristics substantially independ-

ent of the temperature up to the maximum permissible operating temperature. For some other common materials, the lower cut-off frequency is raised appreciably as temperature is increased, whereas the DKT microphone has a lower cut-off frequency below 1 cps at all temperatures. It is reported that the maximum permissible temperature of the DKT microphone is limited to approximately 160 F. It is hoped that a correlation can be established between sound level in the water and pipe-wall vibration or air-borne sound level in the sound chamber. Such a correlation could allow extrapolation to higher temperatures at which the underwater microphone could not be used.

The frequency ranges of both accelerometer and microphone extend to above 10,000 cps. In the testing procedure, frequency analyses are made of the signals from the accelerometer and microphone, in addition to the measurements of

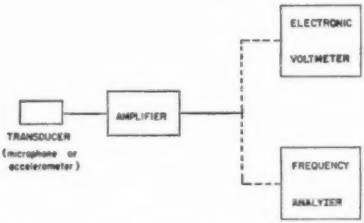


FIG. 3. SIMPLIFIED DIAGRAM OF THE INSTRUMENTATION

magnitudes. Fig. 3 indicates the wiring arrangement of the basic instrumentation.

TEST DATA

Samples of the *types* of data which have been obtained in experimental tests are shown in the accompanying curves. Relative values have been used for the ordinates in presenting these figures.

Fig. 4 illustrates the variation in output signal of the underwater microphone when noises were produced by a 1-in. gate valve as it was used to control water flow. The microphone was placed approximately 3 in. downstream from the valve. Upstream pressure (before the valve) was kept the same for all runs. The microphone position was probably close to a region of intense cavitation. The rise and fall of the output signal corresponded roughly with the audible sounds heard. In another run, the microphone was moved to a position 11 in. downstream from the valve. The curve obtained was practically the same as for the original microphone location.

Fig. 5 shows the pipe-wall vibration for a test similar to that of Fig. 4. For this test, a vibration pickup was placed approximately 1-ft upstream from the valve. The curve shows the same general characteristic pattern as Fig. 4.

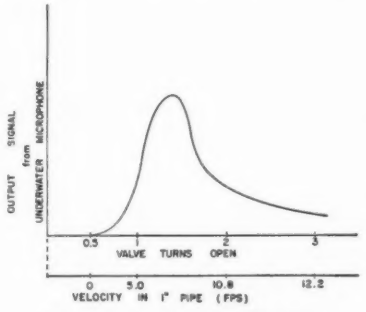
Fig. 6 shows the results of a sample run with constant valve setting and constant flow rate. A frequency analysis was made from the signal produced by a vibration pick-up located approximately 1-ft upstream from the valve. A

particularly strong and rather sharply-peaked component is evident for the particular conditions of this run. The no-flow analysis to establish a datum is also shown.

DISCUSSION AND CONCLUSIONS

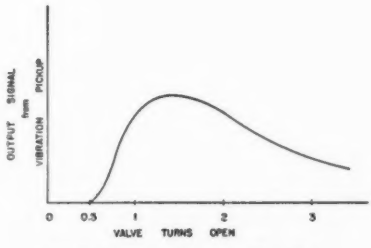
The experimentation to date has been largely of a qualitative nature, mainly concerned with the preliminary testing associated with adjustment of the experimental equipment to give desired operating conditions, and with the determination of proper instrumentation. Although quantitative information is not available at this time, a few generalized conclusions may be of interest.

Cavitation is an important factor, as might be expected. In localized regions of high velocity and low pressure, bubbles of vapor form (or bubbles of gas



One-inch gate valve controlling flow. Constant pressure before valve = 22 psig, temperature = 40-70 F

FIG. 4. RELATIVE MAGNITUDE OF OUTPUT SIGNAL FROM UNDERWATER MICROPHONE AS A FUNCTION OF VALVE OPENING



One-inch gate valve controlling flow. Constant pressure before valve = 26 psig, temperature = 53 F

FIG. 5. RELATIVE MAGNITUDE OF OUTPUT SIGNAL FROM PIPE-WALL VIBRATION PICKUP AS A FUNCTION OF VALVE OPENING

are released), giving rise to hissing sounds. Straight runs of pipe do not appear to contribute significantly to the production of noise. The preliminary tests, which covered a range of velocity up to 10 fps, indicated that when pressure was kept sufficiently high to prevent vaporization or gas release, flow was quiet over the entire velocity range. This conclusion was used in establishing a reference datum with which noises produced by fittings could be compared. Noise generated at valves and fittings far exceeds that for straight pipe runs.

Contractions of flow apparently take place quietly, while sudden expansions or enlargements of the flow stream, and also sharp directional changes, produce noise. In general, more rapid pressure drops are accompanied by increased tendency to generate noise. Obviously the shape of the bounding walls of the flow stream, the pressure, temperature, velocity and probably gas content of the water are significant variables.

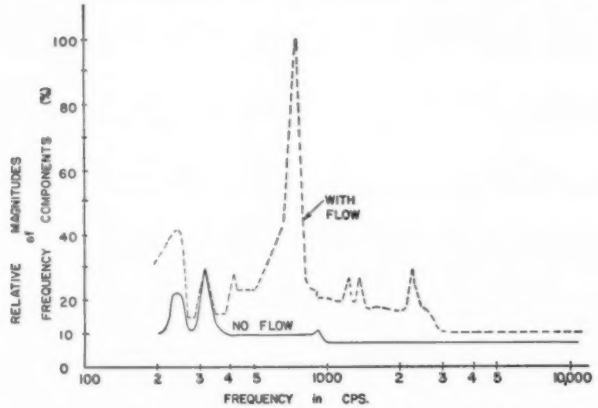
The next phase of the problem is the quantitative determination of the noise and vibration producing characteristics of components of piping and heating

systems. The task is a sizeable one, because of the large number of variables involved.

When a sufficient background of quantitative data has been established against which quieting procedures can be judged, then the evaluation of the quieting effectiveness of various schemes will be undertaken. The frequencies and frequency ranges of noises and vibrations produced will be significant factors in considering means of silencing.

POSSIBLE QUIETING SCHEMES

A few of the schemes which seem to offer possibilities will be surveyed briefly. None of these schemes has been evaluated quantitatively as yet in the



Constant setting of one-inch gate valve at approximately ¾ turn open. Constant velocity of 1.2 fps in one-inch supply pipe. Pressure before valve = 26 psig, temperature = 56 F

FIG. 6. RELATIVE MAGNITUDES OF FREQUENCY COMPONENTS OF SIGNAL FROM VIBRATION PICKUP

course of the current investigation, although some of them have received considerable quantitative attention by others. There is a need for further study of the application and relative effectiveness of the various ideas as related specifically to the problems and the variables encountered by the heating industry.

The remedies must be considered under two general categories: (1) the reduction of the intensity of noises and vibration actually created (the noise sources), and (2) the reduction in the ease of transmission of the noises and vibration to other localities where they could become more troublesome than at the point of origin.

The creation of noises is influenced by the design of the components through which flow occurs (valves, pumps, other fittings), and the manner of operation of the system. As an example of the latter item, if a system is producing noise because of cavitation, it is possible that raising the pressure of the system could reduce or eliminate cavitation and thereby reduce noise. This probably cannot

be considered as a general cure-all because cavitation can be extremely localized and is influenced greatly by the local configuration of the particular fitting or component. Also, the allowable system pressure is limited.

Pressure Drops: As another example, indications are that large, sudden pressure drops tend to produce much more noise than more gradual pressure reductions, and this seems reasonable in view of the high localized velocities and accompanying low pressures which could be produced. The conditions are more favorable to cavitation. It can be deduced that a given total pressure drop should produce less noise if graduated over a series of valves instead of being taken across a single valve. Again, however, the design of the individual valves will influence the situation.

Noise Transmission: Once noises have been produced, there remains the problem of minimizing their transmission. The vibrations which constitute sound are readily transmitted along both the liquid stream and the pipe-wall structure, and can travel long distances with little attenuation. Further, the pressure fluctuations (sounds) which may originate in the liquid very quickly cause vibration of the pipe-wall itself, and so both methods of transmission are of importance. There is apparently rather close coupling between liquid vibration and pipe-wall vibration. This might be expected, because both water and steel are relatively incompressible.

Liquid-Borne Vibrations: The liquid-borne vibrations can be effectively reduced by allowing the liquid to travel through soft rubber hose as has previously been briefly mentioned. The effect is to relieve the pressure peaks as the hose expands, and to dissipate energy by internal damping in the rubber as the hose repeatedly contracts and expands. The method is more effective for high-frequency vibrations than for those of low frequency because more cycles of the higher frequencies will act against the rubber walls in a given length of hose.

Soft-rubber hose has obvious limitations when consideration is given to the pressures and temperatures which must sometimes be encountered, and to the requirements of long-life, trouble-free service. Further, increasing the total liquid pressure results in greater initial distention of the rubber tube, and decreased effectiveness as a sound attenuator. Hose which consists of an inner core of rubber, plus layers of cord or fabric of some sort (such as steam hose) can accommodate reasonable pressures and temperatures for reasonable periods of time, but such hose is noticeably inferior to plain tubing in attenuating sound.

Rubber itself is practically incompressible, and in order to function as a sound attenuator, it must have space into which it can expand as it receives pressure fluctuations. The *give* of the cords around the rubber tube of steam hose is relatively slight, and not much room is available *between* cords, obviously. For comparable effects, the length of steam (reinforced) hose required must be many times that of plain tubing.

Means other than rubber tubing might be used to solve some of the problems. For example, a flow chamber might be packed with small hollow gas-filled spheres with sufficiently elastic walls to allow pressure relief, or with a sponge-rubber type of material with an internal structure of permanently gas-filled cells. The cross-section must be adequate to permit flow without excessive pressure-drop. For high-frequency noises, such a chamber could be relatively small, but for low frequencies it might be prohibitively large. It would be

desirable to have water-to-rubber contact for many cycles (and wave lengths) for effective damping.

It is conceivable that the viscosity of the water itself might be useful if properly utilized. Consider water flow through a porous material in which the water passageways are of capillary size. A chamber packed with some substance such as steel wool is an example. Oscillations of the water would tend to be damped out by the large viscous drag along the bounding surface areas.

Damping by Piping Arrangements: Another possibility for reduction of transmitted noise is offered by suitable piping arrangements which might introduce interference effects. Fig. 7 shows a possible arrangement. A simplified explanation follows. If the length of the branch pipe *B* is such that the part of the sound wave at *A* which traverses the pipe *B* arrives at *C* out of phase with the wave portion which goes straight through to *C*, then a cancellation or interference effect might be expected. Actual sounds encountered will rarely be of single frequency, but such an arrangement might be made to possess somewhat broad-band characteristics. Another related idea is that of the dynamic absorber, in which a system could conceivably be tuned so that a pressure pulse could be

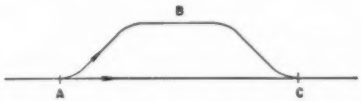


FIG. 7. POSSIBLE PIPE ARRANGEMENT FOR PRODUCING INTERFERENCE EFFECTS

absorbed by an elastic member, and the energy delivered back to fill in the rarefaction portion of the cycle. These schemes remain to be tried.

Isolating Pipe from Structure: To reduce structural-borne vibration, there are the possibilities of isolating the piping from noise (vibration) sources by suitable couplings, and of introducing damping. As a coupling, a suitable length of soft rubber hose can be effective, but there are obvious disadvantages which have been previously mentioned. More rigid couplings which are suitable for higher pressures and temperatures would be expected to be less effective, as they also are in attenuating fluid-borne sound. Damping can be introduced by special pipe supports, pipe wrappings or other devices which allow sliding as vibration occurs. The sliding friction removes energy from the vibration, thereby reducing the amount transmitted.

Another rather obvious item which should be mentioned here is the use of solid pipe supports which effectively transmit pipe-wall vibration to floors and walls of buildings, and sometimes allow large panels to act as sounding boards. Breaking the solid connections by using flexible supports to decouple pipe vibration from the building structure obviously can sometimes be helpful.

This brief survey indicates at least some of the methods and general principles which are pertinent to the problems of this investigation. The various possibilities remain to be quantitatively evaluated for the conditions of interest in the present study. Future reports will present quantitative information which is determined during the continuing course of the investigation.

ACKNOWLEDGMENT

The author wishes to acknowledge especially the active interest, support and cooperation of Prof. B. H. Jennings, chairman of the Department of Mechanical Engineering of Northwestern University Technological Institute, during the inception and continuation of this project.

BIBLIOGRAPHY

1. Eliminating Noise Transmission in Liquid Pipe Lines, by Martin Hirschorn (*Heating, Piping & Air Conditioning,* March 1952, p. 100).

2. Reducing Noise in Machines, by D. B. Calloway (*Machine Design,* December 1951).

3. Noise in Domestic Water Systems, by F. W. McGhan (*Plumbing and Heating Journal,* January 1952).

4. Acoustic Filter for Water-Filled Pipes, by Hoover, Laird, and Miller (*Acoustical Society of America Journal,* January 1950).

5. Resonant Vibrations in a Water-Filled Piping System, by Calloway, Tyzzer, and Hardy (*Acoustical Society of America Journal,* Vol. 23).

6. Propagation of Sound Waves along Liquid Cylinders, by W. J. Jacobi (*Acoustical Society of America Journal,* March 1949).

7. A New Method of Vibration Measurement for the Frequency Range 20 to 20,000 Cps, by Frank Massa (*Instruments,* November 1948).

8. Noise Primer, General Radio Co.

9. Fundamentals of Acoustics, by Kinsler and Frey (John Wiley & Sons, Inc., New York, 1950).

10. Acoustic Measurements, by L. L. Beranek (John Wiley & Sons, Inc., New York, 1949).

DISCUSSION

A. T. JONES, Toronto, Ont., Canada (WRITTEN): This paper is particularly valuable as an introduction to the work being carried out at Northwestern University. It will give the reader a useful reference when studying succeeding papers on this research where for lack of space it will be impossible to include the preliminary data.

It is generally considered that velocities in hot water heating pipe lines should not exceed 4 fps for relatively quiet operation, while some authorities specify 300 milinches per ft as being good practice with a maximum of 400 milinches per ft.

This latter assumption indicates that larger pipes will carry higher velocities than the smaller pipes for relatively quiet operation—for example 400 milinches per ft friction in a ¾-in. diameter pipe corresponds to a 2 fps velocity, while a 4-in. diameter pipe at same resistance per foot has a velocity of 6 fps. It is to be hoped that the final results of this research will solve this problem.

The work to date seems to indicate that velocities as high as 10 or 12 fps might be carried in straight pipe of 1-in. diameter. However, valves and fittings are the troublemakers preventing the use of such velocities in hot water heating systems for residential or office heating. Would it be too implausible to suggest that our whole design of valves and fittings might be greatly influenced by this research?

Screwed valves and fittings might be streamlined to the point where much higher velocities could be satisfactorily employed than at present. The expansion and contraction of the flow when entering or leaving fittings as a result of the inside of

the fitting being significantly larger than the inside diameter of the adjoining piping might even be a source of noise.

In possible quieting schemes the fact that a given valve or fitting has a relatively high magnitude of noise at certain frequencies should be kept in mind.

It is possible that most sounds due to velocity come in a certain range of frequency where a useful quieting scheme might be employed to eliminate the most bothersome component.

In reading this paper three main considerations came to mind: (1) how can noise be reduced to an acceptable level when found in existing systems; (2) what precautions should be taken to design systems for quiet operation using present day fittings; and (3) how can we make use of higher velocities than presently employed while still keeping noise to an acceptable level? No doubt later work will help provide answers to these questions.

R. T. KERN, Fitchburg, Mass. (WRITTEN): This paper starts a study of problems, the answers to which have been needed for a long time by those actively interested in designing hot water heating systems. Although little of practical value has as yet been developed, it is interesting to note that methods of procedure have been set up and now quantitative studies can be started.

There is a great deal of work to be done before practical answers can be expected, but since the noise developed in straight pipe seems to be of very minor importance, it would seem the first studies should be made to find the major causes of noise and the methods than can be used to eliminate them. From personal experience, I feel the major causes of noise are throttling valves and orifices, either or both of which are required on practically all hot water systems larger than residence size.

In most large systems, a certain amount of noise is permissible but the high pitched noise caused by a throttling valve (used to decrease the pressure on the downstream side a considerable amount so as to equalize the friction loss of various circuits) cannot be tolerated. Up to now, the only practical way to reduce this noise is to use multiple throttling valves in series. The number of valves required is almost entirely a guess and the lowest noise level is found by trial and error.

Orifice plates have their limitations also, since they have fixed outlet sizes and the drop through them cannot be changed without removal or change of size, so it would appear we must depend on valves to secure the necessary control. The study of methods that can be used to eliminate the noise they generate should be of prime importance in order to find the most needed answer to the problem.

With the water velocities normally used in heating systems it is my belief that noise generated by fittings of various types is of such low intensity as to be practically negligible. The use of flexible hose of any kind is usually impracticable.

E. L. WEBER, Seattle, Wash. (WRITTEN): There is a growing need for the extensive study of noise elimination in buildings. Great strides have been made by the acoustical treatment industry, but there has been a woeful lack of progress in the heating and ventilating profession. Only in recent years has the noise level of fans been inserted in catalogs.

The noise produced by liquids in pipes and fittings requires much study. Use of pipe bends and welded connections has done much to improve the situation, but valves are still noise producers.

In my experience with large hot-water heating systems, I have found that less noise is produced by using velocities of 10 to 12 fps than is produced by increasing the size of the mains and throttling down the flow with a valve to give the same delivery of water at a lower velocity. The valve increases the noise level.

The velocity of the liquid does not seem to have much effect on the noise level, but the turbulence set up by valves and fittings is the real problem.

In studying air movement noise is produced not by the velocity, but the turbulence set up by improperly designed duct systems and especially by the uneven distribution

over the fan discharge. This requires study and correction by the fan manufacturers.

The valve manufacturers should also endeavor to produce valves, possibly on a venturi tube principle, which will eliminate turbulence and its ensuing noise. A flush valve and flush tank without the usual *sing* just before closing would be particularly helpful.

The use of lead to eliminate transmitted noises opens a wide field for noise level reduction, as the velocity of sound in lead pipe is only one fourth that through iron or steel. Lead has practically no resilience and will hence dampen vibrations.

The studies at Northwestern University can undoubtedly be of great value to the industry, and deserve the support of the Society.

M. W. McRAE, Chicago, Ill.: First, may I say that Professor Rogers has certainly established a very good foundation for proceeding with the quantitative work. It has been a very difficult problem to design and instrument the experimental equipment. His remark that there were many variables involved was an understatement. I have several questions for the author: (1) In the instrumentation, was the use of a rotameter considered to measure the volume of water? Since the water was being weighed, and there was only a 750 gal supply tank employing the batch system, the time is rather limited. Perhaps the use of a rotameter might permit more readings in the short period of time available. (2) Why was a gate valve used in the preliminary experiment? As a gate valve is normally used in the open or closed position, I wondered why any experiments were needed at various gate valve positions, or degrees of opening.

R. T. Kern in his discussion mentioned the limitations of rubber or flexible connections for a heating system. In view of the limitations listed for such connections to suppress the sound transmission, might it not be more advantageous to consider the design of valves or fittings normally used in a heating system?

Professor Rogers is correct in stating that it is possible to utilize the viscosity of the water itself in minimizing the flow noise by providing multiple small passages which result in a large wetted perimeter. Some applications have already been developed along that line to suppress noise in plumbing equipment.

R. C. CHEWNING, Portland, Ore.: This paper is like the early work on physiological research in that it is preliminary. In research on noise in water systems, just as in physiological research, there is a tremendous amount of work required before answers useful to the profession are obtained.

The paper points out that cavitation is the basic cause of noise and that by increasing pressures and eliminating air and other gases as much as possible a quieter system can be obtained. This is of great value, and I think that this information alone pays for the work that has been done thus far.

AUTHOR'S CLOSURE: Mr. Weber mentions the possibility of designing valves using a venturi principle to eliminate turbulence and its ensuing noise. A well-designed venturi which minimizes turbulence will produce only a very small overall pressure drop, and thus probably will not be very effective in controlling flow rate. It would appear that increased irreversibility and turbulence, particularly in the diverging outlet section, would necessarily accompany increased overall pressure drop. The alternative would be pressure recovery in the diverging section. However, it is not inconceivable that such valves might possibly be somewhat quieter than conventional types. Valves are available which make use of the basic venturi idea by squeezing a rubber tube (so-called pinch-type valves). How quiet these valves are, this writer does not yet know.

Mr. Kern indicates that throttling valves and orifices are major sources of noise, in his experience, and that multiple throttle valves in series can reduce total noise. Qualitatively, we agree. It appears that greater rates of pressure drop result in more

intense noise generation. The subject is receiving quantitative attention as the investigation continues.

The three questions asked by Mr. Jones cannot, of course, be answered adequately and specifically at this time.

Perhaps one of Mr. Chewning's statements should be discussed. He states that one fact which had been brought out was that cavitation is *the* basic cause of noise. Cavitation is an important factor, as was stated in the paper, but we would not like to convey the impression that we necessarily believe it to be the *only* cause of noise. Noises accompanying cavitation are apparently relatively intense, but noises conceivably produced without apparent cavitation might also be of importance.

In answer to Mr. McRae's questions, the use of a rotameter was considered, but it was decided that it would be preferable not to include an additional potential noise source in the piping circuit if it could be avoided. Operating experience showed, too, that during the time required for recording the readings of the various instruments, sufficient drop in the tank level was obtained to allow an accurate determination of the flow rate except for the runs at very slow flow rates. There would be very little saving in time if a rotameter were to be used.

In regard to the second of Mr. McRae's questions, there was no logical reason for using a gate valve in some of the preliminary tests. In fact, the focus of attention at the moment was on the characteristics of the instrumentation. It was desired to produce a noise by flow through some piping component which could be expected to produce a relatively intense noise, so that the probable range of readings which would need to be considered could be determined. There was no careful selection of any particular valve. The runs which were made were exploratory for the purpose of checking the required range and other characteristics of electrical and electronic instrumentation.

Several of the discussers mention the possibility of re-design of valves and fittings. Apparently we are all in agreement in thinking that there is no reason to suppose that the usual fittings are of optimum design where noise generation is concerned.

1482

No. 1482

ELECTRICAL ANALOGGER APPLICATION TO THE HEAT PUMP PROCESS

By CARL F. KAYAN*, NEW YORK, N. Y.

REFRIGERANT-VAPOR compression systems, whether for heat-pump or for refrigerating purposes, will operate in steady state at some equilibrium condition determined by relative component performance characteristics. This means that the condensing temperature and the compressor delivery pressure, under fixed evaporator conditions, are the result of comparative performance characteristics of both the condenser and the compressor, as subject to their imposed conditions. Thus, variations in the condenser coolant conditions will be reflected in a variation of the condensing temperature; this, in turn, will affect the resultant compressor performance.

These interdependent relationships for the system may be investigated by electrical Analogger methods, wherein system performance values may be represented by electrical quantities. The use of electrical analogy methods for the study of various steady-state heat-flow systems[1,2,3,4,5] and transient or unsteady-state systems[6] is well known.

For the purpose of clarification and simplification in this initial presentation, many expedient assumptions will be in order and will be made. Idealized performance will be assumed; corrections to specific practical performance of real equipment may readily be made. The present paper is prepared essentially for the purpose of introducing another tool in the electrical Analogger operations. This tool, in general, may serve for the ready determination of overall performance characteristics in the analysis of composite integrated systems comprising fluid machines and associated heat transfer equipment under various imposed conditions and heat load.

GENERAL ASPECTS OF THE VAPOR COMPRESSION CYCLE

The conventional refrigerant-vapor compression system, in terms of a simple component circuit, is illustrated in Fig. 1. Power is supplied to run the compressor; heat is abstracted in the condenser through the medium of a coolant

* Professor of Mechanical Engineering, Columbia University. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

such as water flowing at a given rate; heat is supplied to the evaporator by brine of a refrigerating system, by chilled-water of an air-conditioning system, by atmospheric air flow of an air-source heat pump system, or perhaps by waste warm liquid in an industrial heat pump process.[7]

The pressure-temperature relations for a refrigerant are indicated in Fig. 2, with temperature-pressure levels of system operation shown in Fig. 3, and general corresponding fluid heat-transfer temperatures indicated in Fig. 4. Pressure difference ΔP and the corresponding temperature difference Δt effected

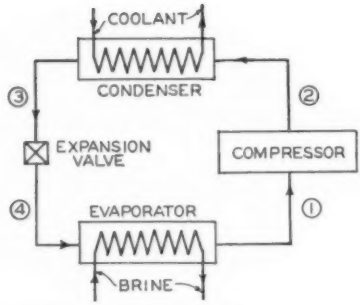


FIG. 1. SCHEMATIC LAYOUT OF VAPOR-COMPRESSION CIRCUIT

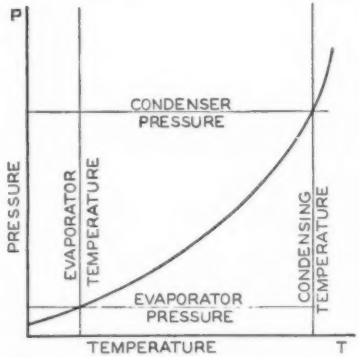


FIG. 2. TYPICAL PRESSURE-TEMPERATURE CURVE FOR REFRIGERANT, WITH CYCLE TERMINAL CONDITIONS INDICATED

by the thermodynamic system are likewise indicated in Fig. 3. The pressure-enthalpy (P-h) diagram for an ideal refrigerant cycle is shown in Fig. 5 with particular reference to component enthalpy differences Δh, *i.e.*, for evaporator, compressor and condenser.

COMPRESSOR PLANT

For the purposes of this paper a positive displacement compressor system with the usual machine suction and discharge valves is assumed. Equivalent analysis, however, could be carried out with other types of compressor systems. The performance of the compressor depends on its construction features, *i.e.*, the ideal displacement gas volume is modified by the effect of the overall volumetric efficiency. Clearance in conjunction with pressure ratio, pressure drops in valves, suction gas heating, as well as the specific gas properties, directly affect the effective gas handling capacity, as measured by *real* volumetric efficiency. On an idealized basis, without considering pressure drops through suction and delivery valves, and suction heating, the volumetric efficiency may be shown as:

$$E_v = 1 + C - C\,(P_2/P_1)^{1/n} \quad \ldots\ldots\ldots\ldots \quad (1)$$

The ideal volumetric efficiency, though itself less than 100 percent, is still further reduced by the practical effects of valve drops and heating of suction

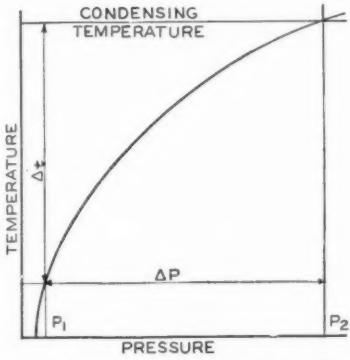


FIG. 3. CYCLE TEMPERATURE-PRESSURE CURVE

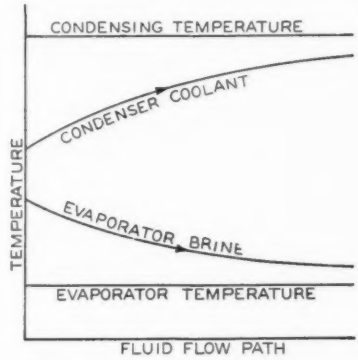


FIG. 4. TYPICAL SYSTEM HEAT-TRANSFER TEMPERATURE RELATIONSHIPS

gas. Fig. 6 shows the general variation of the hourly mass of gas handled *vs.* gas delivery pressure, for $E_v = 100$ percent (unimpaired displacement), ideal volumetric efficiency as given by Equation 1, and the real volumetric efficiency of a practical machine. Fixed suction pressure is assumed.

VAPOR CYCLE

Based on fixed suction conditions such as suction pressure and suction gas quality, and variable discharge pressures, the ideal compressor work per pound of refrigerant (at constant entropy) may be calculated. Similarly, based on

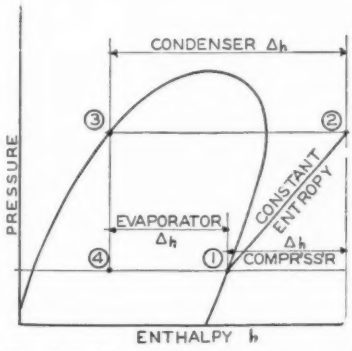


FIG. 5 *(left)*. PRESSURE-ENTHALPY RELATIONSHIPS FOR TYPICAL REFRIGERANT CYCLE

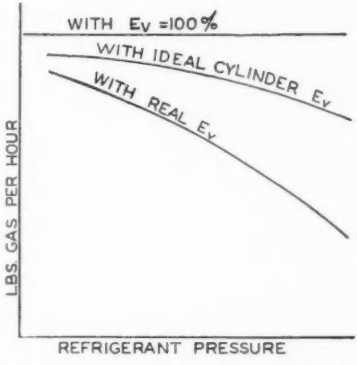


FIG. 6 *(right)*. TYPICAL VARIATION OF COMPRESSOR GAS PUMPING RATE WITH REFRIGERANT DISCHARGE PRESSURE, FOR DIFFERENT VOLUMETRIC EFFICIENCY CONDITIONS, CONSTANT SUCTION PRESSURE

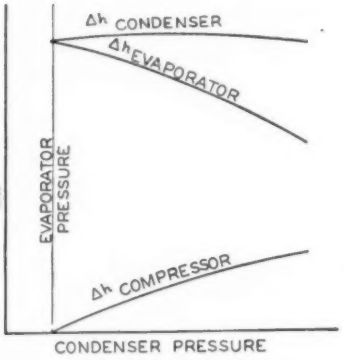


FIG. 7. TYPICAL VARIATION OF COMPONENT ENERGY VALUES PER POUND WITH CONDENSER PRESSURE, CONSTANT EVAPORATOR PRESSURE

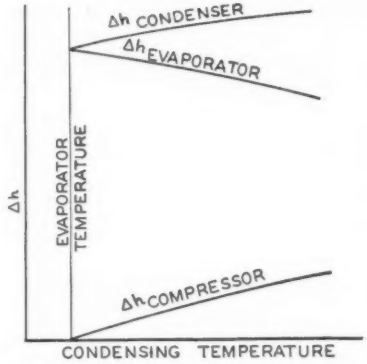


FIG. 8. TYPICAL VARIATION OF COMPONENT ENERGY VALUES PER POUND WITH CONDENSING TEMPERATURE, FOR CONSTANT EVAPORATOR TEMPERATURE

the assumption of liquid refrigerant leaving the condenser at the temperature corresponding to the condensing pressure and entering the pressure drop device (expansion valve) at the same condition, values of the heat per pound given up in the condenser may likewise be calculated; and also for the heat absorbed in the evaporator. Such values for a fixed evaporator pressure are shown plotted

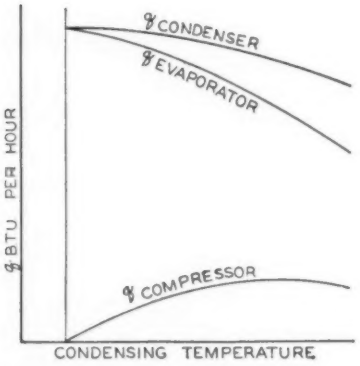


FIG. 9. VARIATION OF COMPRESSOR ENERGIES PER HOUR WITH CONDENSING TEMPERATURE, FOR CONSTANT EVAPORATOR TEMPERATURE AND INCLUDING COMPRESSOR VOLUMETRIC EFFICIENCY EFFECT

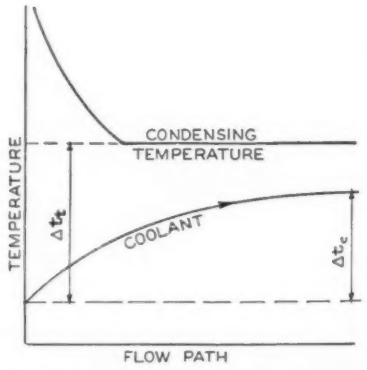


FIG. 10. CONDENSER HEAT-TRANSFER TEMPERATURE VARIATION VS. COOLANT FLOW PATH

against cycle condensing pressure in Fig. 7, and against corresponding condensing temperature in Fig. 8.

Assuming the values of pounds gas pumped per hour shown in Fig. 6 are based on real volumetric efficiency, and combined with the data of Fig. 8, equivalent hourly heat values representing practical performance are obtained and shown in Fig. 9. It is immediately evident that the heat to be rejected in the condenser is made up of that acquired in the evaporator plus that contributed by the work effect accomplished in the compressor.

APPLICATION OF THE RESISTANCE CONCEPT

Condenser Heat Transfer: Disregarding the effect of super-heat on condenser performance, the flow of heat in a refrigerant condenser may be considered to

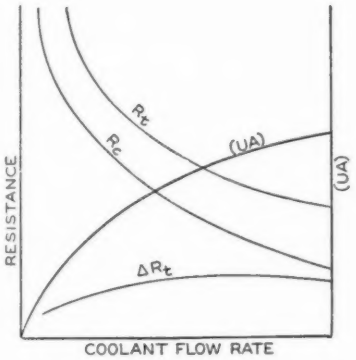


FIG. 11. TYPICAL VARIATION OF CONDENSER THERMAL RESISTANCES AND *UA* VALUE WITH COOLANT FLOW RATE

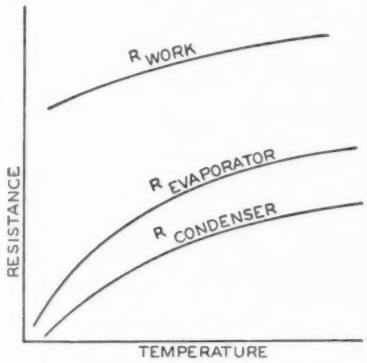


FIG. 12. TYPICAL VARIATION OF COMPRESSOR COMPONENT-ENERGY RESISTANCES WITH CONDENSING TEMPERATURE, FOR FIXED EVAPORATOR TEMPERATURE

vary with the temperature difference between the condensing temperature and the inlet coolant temperature, with fluid temperature relationships as shown in Fig. 10. As indicated in an earlier paper,[5] the hourly heat transfer may be shown in terms of resistance:

$$q = \Delta t_t / R_t \quad \ldots \quad (2)$$

$$R_t = \frac{R_c}{\left(1 - 1/e^{R_c/R_{UA}}\right)} \quad \ldots \quad (3)$$

$$R_c = 1/(Wc) \quad \ldots \quad (4)$$

$$\Delta R_t = R_t - R_c \quad \ldots \quad (5)$$

These relationships are shown on a general basis in Fig. 11, with values plotted against coolant flow rate, W.

Energy Transfer in Machines: Considering the *energy relationships* for a compressor system as shown in Fig. 9, just as the *resistance concept* has been ex-

NOMENCLATURE

A = heat transfer surface area, square feet.
C = clearance fraction, based on displacement.
c = coolant mean specific heat, Btu per (pound) (Fahrenheit degree).
E_v = volumetric efficiency.
n = exponent in Pv^n = Constant (ideally $n = k = c_p/c_v$.)
P = pressure, pounds per square inch, absolute.
ΔP = pressure difference between evaporator and condenser, pounds per square inch.
P_1 = intake pressure of compressor, pounds per square inch.
P_2 = delivery pressure of compressor, pounds per square inch.
q = heat energy as applied to different elements of the system and identified by subscripts, Btu per hour.
R_e = heat exchange resistance, thohms = Fahrenheit degrees per (Btu per hour).
R_t = equivalent thermal resistance for condenser process, Fahrenheit degrees per (Btu per hour).
R_{UA} = net heat transfer resistance, thohms = $1/(UA)$ Fahrenheit degrees per (Btu per hour).
r = electrical resistance corresponding to thermal ohms.
Δt = temperature difference between evaporator and condenser saturation values, Fahrenheit degrees.
Δt_1 = temperature difference between condensing temperature and inlet coolant temperature on condenser, Fahrenheit degrees.
U = overall heat transfer conductance, Btu per (hour) (square foot) (Fahrenheit degree).
W = pounds coolant per hour.

tended to cover heat exchanger transfer due to over-all temperature difference,[5] so may it similarly be extended to cover energy flow in a machine, as for example, in a compressor. Equivalent resistance values may be used to represent the compressor performance of the present system. Assuming a fixed evaporator pressure and corresponding temperature, and a variation of compressor performance in terms of the overall temperature difference on the system, resistance values may be obtained by dividing the temperature difference by hourly heat quantities:

$$R = \Delta t/q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Fig. 12 shows resistance values plotted against temperature difference, as follows:

R_{evap} = resistance value corresonding to hourly evaporator heat.
R_{cond} = resistance value corresonding to hourly condenser heat.
R_{work} = resistance value corresonding to hourly compressor work.

ELECTRICAL ANALOGGER CIRCUIT FOR SYSTEM STUDIES

Using electrical resistance values proportional to the thermal resistances cited in Figs. 11 and 12, an electrical Analogger circuit may be arranged to facilitate ready evaluation of heat pump or refrigeration system performance under different operating conditions. Although the use of other driving forces such as enthalpy differences could be considered, temperature difference seems to offer the greatest advantage, with electrical voltage as its counterpart. Furthermore, as far as the compressor system is concerned, although a variable voltage electrical generator might be employed, the simplest approach appears to be to use an impressed overall voltage as in the case of the condenser system.

It is apparent that whatever heat is discharged from the compressor system must in turn flow through the condenser system. The temperature difference that the condenser heat flow requires under the condenser fluid condi-

tions must in turn set the condensing vapor temperature; this in turn determines the corresponding compressor delivery pressure. The actual compressor system energy, depending on real volumetric efficiency and hourly displacement, is directly dependent on the delivery pressure.

In setting up the Analogger circuit, separate adjustable resistances, in parallel, and representing the combined compressor work and the evaporator phases, are employed. The sum-total current contributed by these two branches must in turn equal the total flowing in the condenser circuit, with the same top potential in effect for both phases. Potential level corresponding to the condenser inlet coolant temperature is maintained on the condenser circuit, using an adjustable ballast resistance. (If, beyond the condenser, the evaporator circuit is to be detailed separately, there must be an additional parallel circuit whose flow must obviously correspond to that for the work-component on the compressor side. Thus the circuit below the condenser would have two branches.)

Fig. 13 shows the simulation circuit, with resistances for R_c and ΔR_t adjustable in accordance with coolant flow rate derived from a working curve such as Fig. 10 for the specific compressor-matching condenser of the system. Fluid temperature corrections may be introduced into Fig. 10 if desired. Also shown in Fig. 13 are resistances for the compressor-work and the evaporator-energy components, adjustable to keep step with the overall value of the compressor-plant potential, in conformance with the appropriate compressor-plant performance characteristic as shown in Fig. 12.

Adjustment of the circuit is made by changing the top-side potential value through the variable resistance r_v, such that the condenser current I_{cond} equals the total compressor-energy current I_{comp}, and with parallel minor corrections made in r_{work} and r_{evap} in accordance with the operating overall potential differences (Fig. 12). The effect of the coolant entering temperature of a given value is achieved by maintaining the potential at point D at proper value. Coolant outlet temperature is determined by potential measurement at point C on the condenser circuit. The various current meters show the equivalent energy flowing at any given point. Adjustment of the different elements of the Analogger for this type of heat-power system is a matter of manual dexterity, somewhat simplified by the use of automatic equipment, such as, for example, a voltage regulator.

Example: Figs. 14 and 15 show data for a given vapor-compression system, with ammonia as the refrigerant and the evaporator operated at a fixed temperature of 0 F. With allowance for *real* volumetric efficiency as illustrated in Fig. 6, hourly heat values as typically illustrated in Fig. 9 are shown in Fig. 14. Corresponding resistance values as calculated according to Equation 6 are likewise shown on the diagram, as *thohms*. Fig. 15 shows values for the condenser typified by Fig. 11, and calculated according to Equations 4 and 5. Values for the product UA are included, all values being plotted against hourly rate of condenser coolant (water) circulation. (Compressor displacement, cfm = 37.3; condenser heat transfer surface = 56.2 sq ft).

Circuit values, as translated from Figs. 14 and 15 for the arrangement of Fig. 13, are on the basis of 1 ohm electrical resistance = 10^{-6} thohms, 0.10 volt = 1 F; 0.100 amp = 1,000,000 Btu per hr.

Fig. 16 shows the variation of condenser heat, in Btu per hr, *vs.* inlet water temperature, with a fixed flow rate of 6,000 lb water per hour for the condenser with evaporator at 0 F.

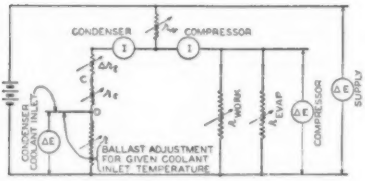


FIG. 13. ELECTRICAL ANALOGGER CIRCUIT FOR HEAT-PUMP PROCESS

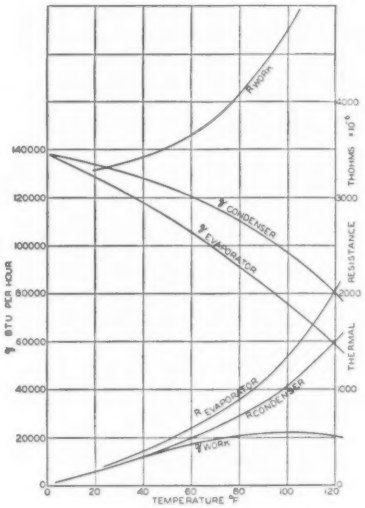


FIG. 14. VARIATION OF COMPRESSOR SYSTEM COMPONENT ENERGY RATES AND EQUIVALENT RESISTANCES WITH CONDENSING TEMPERATURE, FOR FIXED EVAPORATOR TEMPERATURE (NUMERICAL EXAMPLE)

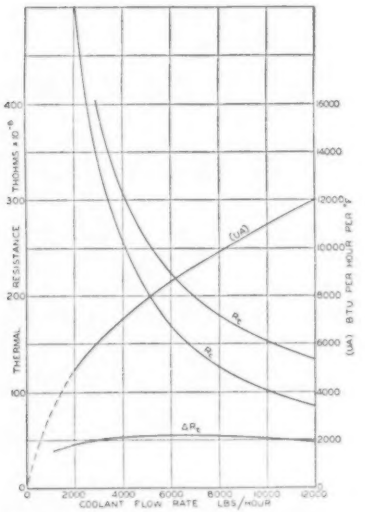


FIG. 15. VARIATION OF CONDENSER *UA* AND CORRESPONDING THERMAL RESISTANCES WITH COOLANT FLOW RATE (NUMERICAL EXAMPLE)

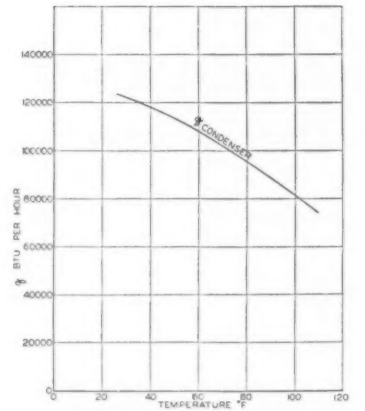


FIG. 16. VARIATION OF CONDENSER HEAT PER HOUR WITH INLET WATER TEMPERATURE (NUMERICAL EXAMPLE)

ACKNOWLEDGMENT

Acknowledgment is gratefully made to associates of the author who have co-operated in various aspects of this study, particularly Prof. J. A. Balmford of the Electrical Engineering Department, and Messrs. P. R. Ellard and D. W. Blair of the Mechanical Engineering Department, Columbia University.

REFERENCES

1. An Electrical Geometrical Analogue for Complex Heat Flow, by C. F. Kayan (*A.S.M.E. Transactions,* November 1945, p. 713).

2. Effect of Floor Slab on Building Structure Temperatures and Heat Flow, by C. F. Kayan (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 377).

3. Temperature and Heat Flow for a Concrete Slab with Imbedded Pipes, by C. F. Kayan (*Refrigerating Engineering,* Vol. 54, August 1947, p. 143).

4. Fin Heat Transfer Analysis by Electrical Analogy, by C. F. Kayan (*Industrial Engineering Chemistry,* Vol. 40, 1948, p. 1044).

5. Heat Transfer Equipment Analysis by the Resistance Concept, by C. F. Kayan (*Refrigerating Engineering,* December 1951).

6. Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 75).

7. Heat Pumps in Heat Recovery, by C. F. Kayan (*Chemical Engineering,* Vol. 57, May 1950, p. 146).

DISCUSSION

E. R. AMBROSE, New York, N. Y.: An electric Analogger circuit as described by Professor Kayan can be advantageously used to evaluate the performance of a heat pump or refrigeration system only when a considerable number of different operating conditions are to be expected. The time and effort required to make the set-up would otherwise make such an analysis impractical. There are, however, many possibilities for its use and the information contained in the paper is a valuable contribution.

W. F. FRIEND, New York, N. Y.: As a member of the Joint *A.E.I.C.-E.E.I.* Heat Pump Committee, which through some seven years has been working to improve the characteristics of the heat pump, several questions have occurred to me.

Would Professor Kayan indicate how he could introduce into the Analogger the important factor of liquid subcooling? On one of the charts is the term *Thohm;* many of the listeners are unfamiliar with what that means in relation to voltage, current or the other characteristics of electric circuits. In the text, some of the calculations have been based on the properties of ammonia. What was the reason for selecting it, rather than dichlorodiflouromethane (F-12)? Have any prototype Analoggers been built so far, and if so, could the approximate cost of such an instrument be given?

Again, the figures are based on the water-to-water type of heat pump, whereas, the large majority of installations, particularly in the smaller or so-called *package unit* class, are air-to-air. To what extent would that kind of unit complicate the preparation of the characteristic resistance curves which represent the data which you have to feed into the Analogger, and the analysis itself? Obviously, in working through the details of this paper, some of the more important potential applications

and certain features which might have more importance than others, probably come to mind. Would Professor Kayan care to suggest one or two of those?

Finally, does the author have in mind how we might elaborate on the arrangement which so far deals only with Btu and temperatures and the other physical properties, in order finally to obtain some information on economic considerations? For example, it might be feasible to draw up curves which show the variation in cost of evaporators or condensers in relation to physical size or thermal performance. Conceivably, we might in one way or another put all of these economic elements together and come up with a family of curves which would show the optimum application of compressor size, evaporator size, condenser size, and, of course, the extent to which the system is influenced by liquid subcooling.

AUTHOR'S CLOSURE: The matter of liquid subcooling could readily be introduced into the system. The heat rejection phase of the cycle would be represented by a condenser plus a heat exchanger, liquid-to-liquid or liquid-to-gas, with the two tied together. Each one would have to be represented by its own thermal resistance set-up. This could easily be done, and would be a further refinement.

On the mysterious designation *Thohm*, it might be said that those of us who have worked with thermal resistances for many years have had to be content just to say *thermal resistance*. The concept of thermal resistance is only a thermal version of Ohm's law. After waiting in vain for a better mind to present some word to designate thermal ohms, the author about ten years ago proposed the combination *thermal-ohm* or simply *thohm*. This term has not been questioned and no one has proposed a different word that could better represent the idea of resistance in the treatment of thermal circuits.

Although ammonia is used as the refrigerant in this paper, it is not a requisite; any refrigerant may be employed. The large municipal heat pump installation at Zurich, Switzerland, which was viewed just four days ago, is an ammonia installation and is of more than passing interest. However, the present paper could equally well have been developed in terms of methyl chloride, or any of the commercial refrigerants.

It should be noted that the analysis could likewise have been carried out on an air-to-air basis: there would be a refrigerant circuit, and the evaporator could be operated with air as the heat source in place of a liquid source. In turn, with the condenser there would be pounds of air entering instead of pounds of water—equally applicable and presenting no special problem.

The Analogger, through a meter or some form of digital indicator could show the cost of the evaporator, or condenser, or any other economics item. This could not be done in its present form, but the various operations could be carried out so as to furnish data whereby working curves of all types could be established. That is the advantage of the Analogger. It is not well adapted for a single computation, as pointed out by Mr. Ambrose; it would not be worthwhile in view of the large amount of preparatory calculations required, as shown on the last charts displayed. The working curves could readily lead the way to economic interpretations.

1483

No. 1483

DESIGN AND PERFORMANCE OF A RESIDENTIAL EARTH HEAT PUMP

By MERL BAKER*, LEXINGTON, KY.

IN SELECTING a heating system for a residence, the owner is usually vitally concerned with (1) dependability, (2) flexibility, (3) cleanliness, (4) cost of installation, and (5) operating cost. He would like to receive maximum advantage of the first three factors and simultaneously keep the latter two as low as possible. In order to select the best system for a particular buyer, the heating engineer is confronted with a difficult problem in that local conditions may greatly influence the desirability of his choice.

The factor of cleanliness was very important in choosing a heating plant for the residence reported in this article. Coal-fired equipment was eliminated as the owner felt that soot would be detrimental in defacing the white shingles of the new house. Natural gas was not available at the location and predicted installation and operating cost for oil-fired equipment was unfavorable. The operating cost of an electric resistance system was discovered to be extremely great even considering the very satisfactory electrical rate in the locality.

The use of a heat pump was considered because no one conventional system seemed to meet the heating requirements satisfactorily. An analysis proved favorable in every respect except for the initial cost. A further consideration revealed that since the heat pump could both heat and cool the house, it would be justifiable to charge only a part of the initial cost to heating and to allow the remainder to be absorbed as the luxury of cooling. Accordingly, the heat pump was selected as the most desirable system for the application. The next problem confronting the designer was the choice of the heat source, with the possibilities being well water, the earth, and atmospheric air. The latter source was eliminated when a check of U.S. Weather Bureau records revealed that during the past 10 years there had been several days with a minimum temperature between — 10 and — 20 F. As a water well did not exist on the property, an embedded earth coil was selected as the most suitable heat source for this particular heat pump.

DESCRIPTION OF INSTALLATION

Conditioned air is supplied to the rooms of the house (shown in Fig. 1) by a forced circulation system. The air conditioning coil transfers heat between

* Associate Professor of Mechanical Engineering, University of Kentucky. Junior Member of A.S.H.V.E.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

FIG. 1. THE RESIDENCE IN WHICH THE HEAT PUMP IS LOCATED

the circulating air and the charge (dichlorodifluoromethane) of the refrigeration system, and serves as a condenser for winter operation and as an evaporator for summer operation. The other major heat exchanger of the refrigeration system consists of five coils of 5/8-in. O. D. tubing connected in parallel and immersed in a 1000-gal storage bath of antifreeze solution (Fig. 2). By use of appropriate conversion valves the immersed coils serve as an evaporator and condenser for winter and summer operation respectively. Heat taken from or given to the storage bath is ultimately transferred to the earth by circulating the fluid through an embedded coil.

The earth coil consists of three parallel circuits of 7/8-in. O.D., type K copper tubing with an aggregate length of 480 ft. The average depth of burial is 6 ft below the earth's surface, and the spacing between coil sections is approximately 8 ft. Figs. 3 and 4 show, respectively, the antifreeze circulation pump and thermometer wells in the headers at entrance to, and exit from, the earth heat exchanger.

The refrigeration compressor, 5 hp driving motor, 1/4 hp circulating fan, air conditioning coil, conversion valves, and auxiliary refrigeration equipment are located within a closed plenum chamber shown in Figs. 5 and 6. The fan draws air at room condition into the chamber through the return duct system and discharges it through the conditioning coil into the distribution system shown by Figs. 7 and 8. Conversion from winter to summer operation is achieved by means of a valve system which reverses the function of the two major heat exchangers. The circuit diagram of the refrigeration system is shown in Fig. 9.

OPERATING CONDITIONS

The nine-room well-insulated house, in which the heat pump is installed, contains a volume of 16,000 cu ft, and has a heat loss of 48,000 Btu per hr based on 70 F inside and — 5 F outside. It is located at Hopkinsville in southwestern

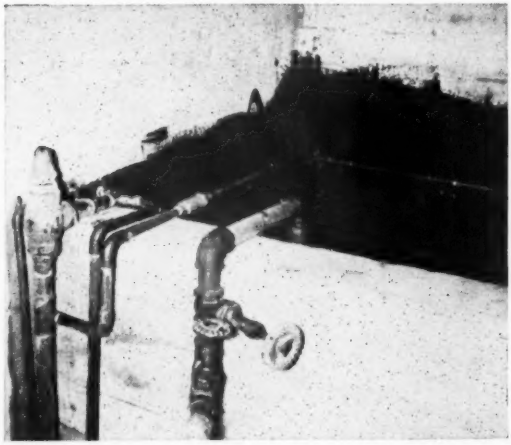


FIG. 2. ANTIFREEZE STORAGE TANK AND DIRECT EXPANSION COILS

Kentucky, approximately 37 deg N latitude and 524 ft elevation. The annual mean temperature for the locality is approximately 55 F, with expected summer and winter extremes of 100 F and — 10 F respectively.

The equipment was combined to provide a refrigeration unit having a rated heating capacity of 53,000 Btu per hr at an evaporator temperature of 10 F,

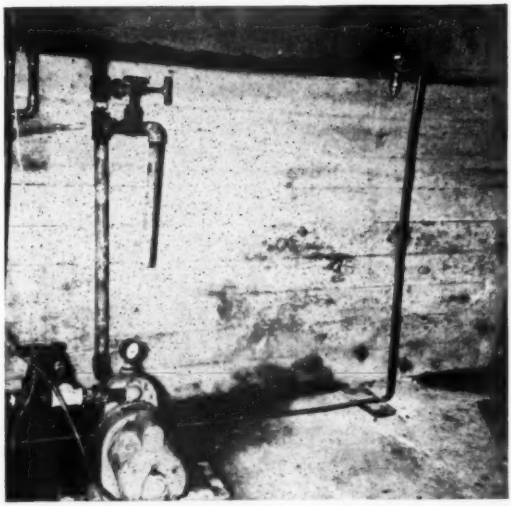


FIG. 3. ANTIFREEZE CIRCULATING PUMP

FIG. 4. THERMOMETER WELLS IN THE ANTIFREEZE LINES ENTERING AND LEAVING EARTH COIL

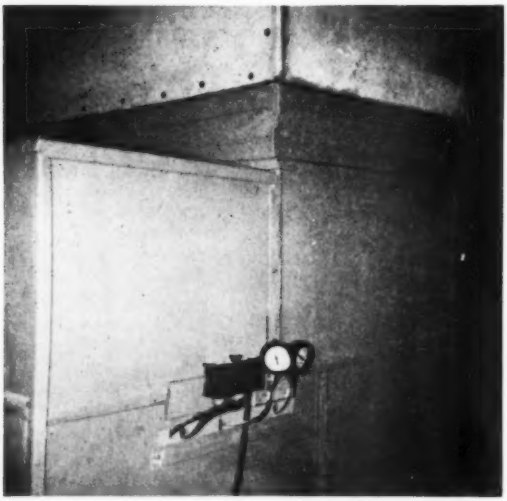


FIG. 5. HEAT PUMP UNIT TOGETHER WITH PRESSURE CONTROLS AND GAGES

and a supply air temperature of 105 F. It is evident that if the coil length were increased to allow adequate heat absorption at a higher evaporator temperature, the required displacement of the compressor would be less. An evaporator temperature of 10 F is a design value only. Operation normally proceeds at considerably higher temperatures. The temperature of the supply air was determined by an economic consideration of the first cost of the fan and air conditioning coil and the operating cost of the refrigeration plant as influenced by head pressure.

The thermal capacity of the 1000-gal antifreeze storage bath permits operation at a high average suction temperature, a factor conducive to good efficiency.

FIG. 6. HEAT PUMP UNIT SHOWING SOME OF THE CONVERSION VALVES

A time delay switch allows the circulation pump to operate longer than the compressor thereby reducing the instantaneous rate of heat transfer between the antifreeze and earth. Only during continuous operation at the design load would the rate of heat transfer between the antifreeze and earth be as great as that between the refrigerant and the antifreeze.

The air temperature within the house was controlled by a central thermostat located in the first story and connected in series with the starter coil of the compressor motor. Room temperatures were satisfactory. Since only two people normally occupied the house, registers to two of the rooms were sometimes closed. The heat actually supplied by the plant during the 1950-51 heating season was only 63 percent of that estimated by the degree-day method. The decrease below calculated values, while partially due to the reduction in heat supplied to two of the rooms, was also due to snow on the roof, Venetian blinds, and good weather stripping.

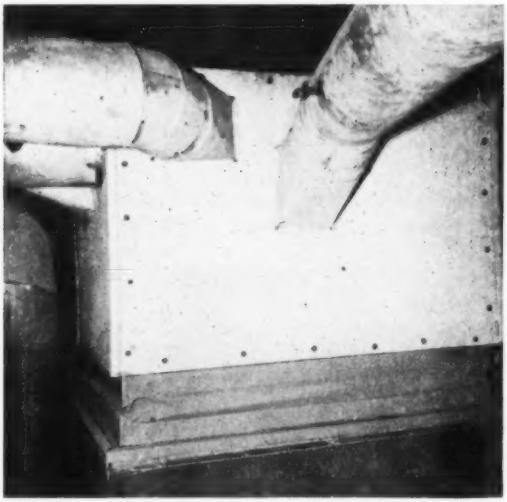


FIG. 7. PLENUM CHAMBER AND DUCTS

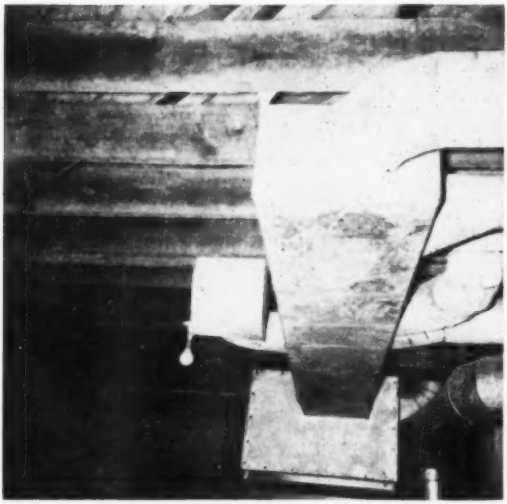


FIG. 8. PLENUM CHAMBER AND TRUNK DUCT SYSTEM

PERFORMANCE DURING 1950-51 HEATING SEASON

An electric clock was connected in parallel with the compressor motor to indicate daily running times. The plant ran most frequently between 10 p.m. and 9 a.m., thus producing a favorable load as viewed by the electric company. An electric power meter was used to measure the combined input to the compressor and fan motors. By recording the suction and discharge pressures, the hourly rate of heat absorption by the evaporator was determined from the manufacturer's data. The product of this rate and the actual operating time added to the electrical input gave the heat delivered to the building. Radiation and convection loss from the air distribution system were not charged against the

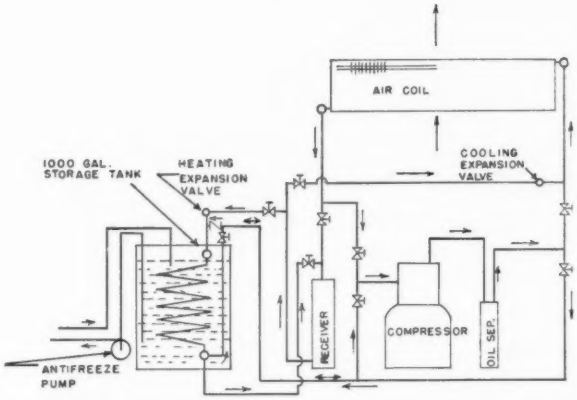


FIG. 9. DIAGRAM OF REFRIGERATION CIRCUIT

heat pump. The daily time of operation throughout the season is shown in Fig. 10.

Calculations indicated that the 1000-gal evaporator storage tank would have to be supplied with sufficient antifreeze to enable operation at a temperature of approximately 15 F. Water was used as late in the heating season as possible in order to extend the test period for tank leaks. About mid-November, ice was forming on the outside of the immersed coils while the temperature of the water in the tank ranged from approximately 45 to 55 F, fluctuating with the operative cycle. The ice, however, readily disappeared when the compressor was stopped and the build up during the running cycle was not significant. On November 25, when the outside air temperature dropped to — 4 F, ice formation required stopping of the plant for a few minutes to allow de-icing of the coils, after which normal operation was resumed. The installation of a larger capacity circulating pump, running longer than the compressor minimized ice accumulation and, consequently, operation for the remainder of the winter was achieved with only water in the tank. On a few extremely cold days the pump ran continuously. The mean water temperature of the tank throughout the winter is shown in Fig. 11, which shows that a minimum temperature of 36 F was reached about

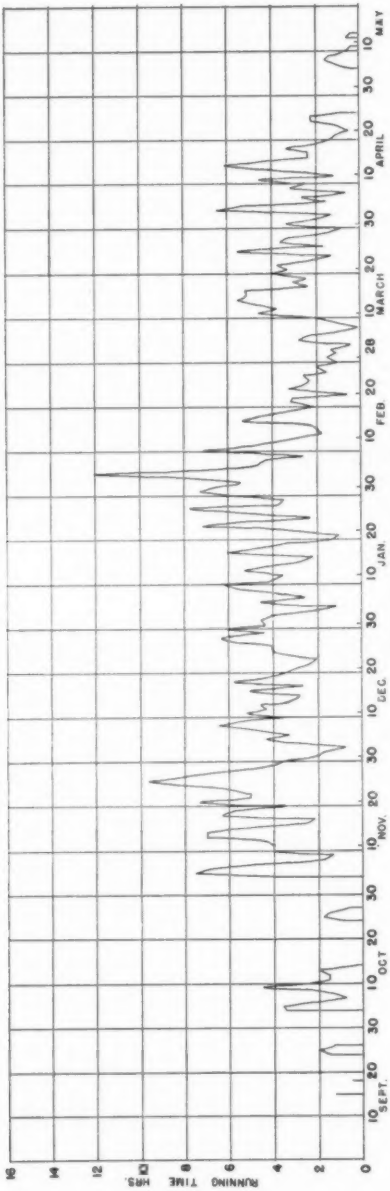


FIG. 10. OPERATING TIME OF UNIT

February 8 after which it began to rise. Ice formation was the most severe during the early part of February while the tank bath was the coldest, but even then the insulating effect rarely forced the suction temperature of the refrigerant below 17 F.

By observing the temperature of the water returning from the earth coil at the start of the circulating pump (measured by a mercury thermometer inserted within a thermometer well in the return line) an approximation of the soil recovery temperature was obtained. Table 1 shows the comparison between the approximate recovery and the calculated undisturbed soil temperature, together with the average heat abstraction during the period. The favorable

TABLE 1—EARTH HEAT EXCHANGER DATA, 1950-51

MONTH	AVE. HEAT ABSORBED, (Btu/(hr) (ft)	OBSERVED APPROX RECOVERY TEMP F	CALCULATED UNDISTURBED TEMP F
September 1950	1.7	63	64
October 1950	3.7	61	62
November 1950	17.6	58	58
December 1950	15.4	52	54
January 1951	16.4	44	49
February 1951	12.3	42	46
March 1951	10.7	44	46
April 1951	7.8	46	48
May 1951	1.5	50	52

comparison between the recovery temperature and calculated undisturbed temperatures is of great value to the designer. Precipitation above normal occurred during the year and much of it was in the form of snow. The prediction of the insulating effect of the snow which began in November and extended intermittently into April was difficult. During a considerable portion of the period the ground was blanketed with snow to depths ranging up to 1 ft. Comparative performance data and weather conditions are shown in Fig. 11 and Table 2.

DISCUSSION OF TEST DATA

The somewhat limited data reported in this article are not intended to represent a complete engineering test, but to serve the important role of evaluating the performance of a heat pump system under actual operating conditions. From the data presented, the performance of the earth heat exchanger may be evaluated accurately together with the efficiency of the plant and cost of operation. A discussion of a complete engineering test is given in previously published material.[1,2]

The results of the 1950-51 heating season test indicate that the earth heat exchanger was adequately sized for the demand and that the short term maximum heat abstraction rate (determined from the design heat loss) is the critical design load factor. However, at abstraction rates considerably greater than

[1] Exponent numerals refer to References.

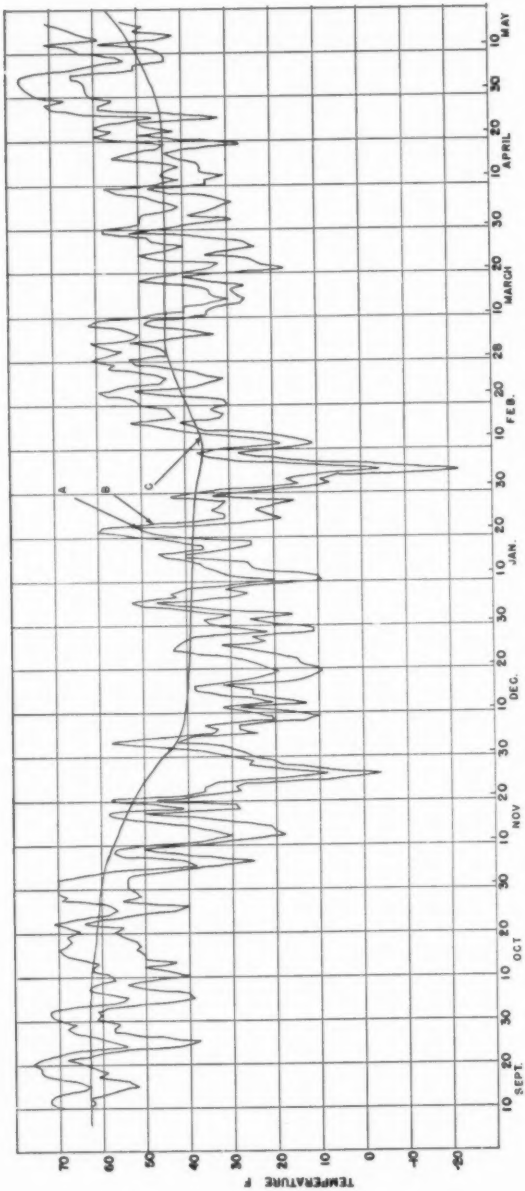


FIG. 11. TEMPERATURE OF OUTDOOR AIR AND STORAGE BATH

(Temperatures: A—Minimum Outdoor, B—Mean Outdoor, C—Storage Tank)

encountered in this test, the progressive decrease in the temperature of the earth produced by the continuous removal of heat may prove more significant. A recovery record similar to the one reported in this analysis, Table 1, was observed by Penrod and associates for an approximate average absorption rate of 15 Btu per (hr) (ft) and a cycle of 12 hr operative and 12 hr idle.[3] From their report maximum recovery temperatures from November 1 through May 1 for the tube surface and a point 18 in. below were:

MONTH	TEMP OF TUBE SURFACE F	TEMP POINT 18 IN. BELOW TUBE (6 FT BELOW SURFACE) F
November 1	63	63
December 1	51	55
January 1	46	50
February 1	43	43
March 1	42	46
April 1	42	44
May 1	44	46

The favorable comparison between results of short term and seasonal tests was also reported.

The Ingersoll-Plass, the Binder-Schmidt and the relaxation methods, all approximate, are very valuable in predicting the performance of a coil embedded in the earth. An observation of the slope of the temperature distribution curve in the proximity of the coil for the Binder-Schmidt representation, Fig. 12, emphasizes the importance of knowing the correct thermal conductivity of the soil. A significant decrease of this property under that used for design would prove very detrimental. Results of the present test indicated that the actual value of the thermal conductivity k was higher than that assumed, but quantitative evaluation was impossible in the absence of measured temperature distributions around the coil. Because of the extreme importance of knowing the correct value of k, a soil analysis is invaluable. This factor is discussed in Reference 3.

It is well to mention that ice formation next to the embedded tube will aid in increasing the apparent thermal conductivity in this region where an increased conductance is most needed. The latent heat supplied from freezing will produce a favorable effect, but is of very limited duration. This is seen by observation from the Binder-Schmidt analysis, Fig. 12, in that the freezing temperature advances only 0.2 ft around the tube in the 12-hour period investigated. As chilled water was used throughout the complete test reported, no advantage was realized from freezing the earth. However, heat absorption by the chilled water from the two sides and bottom of the storage tank exposed to the earth and from rather long connecting headers did aid the 480 ft of coil in supplying the load. Design of the earth heat exchanger is probably the most important factor affecting the overall performance of a heat pump. Insufficient area necessitates the use of a large displacement compressor and results in low efficiency because of the low temperature at which the refrigerant must be evaporated in absorbing the necessary heat energy. Oversizing, on the other hand, may be just as detrimental in that excessive coil surface can easily make

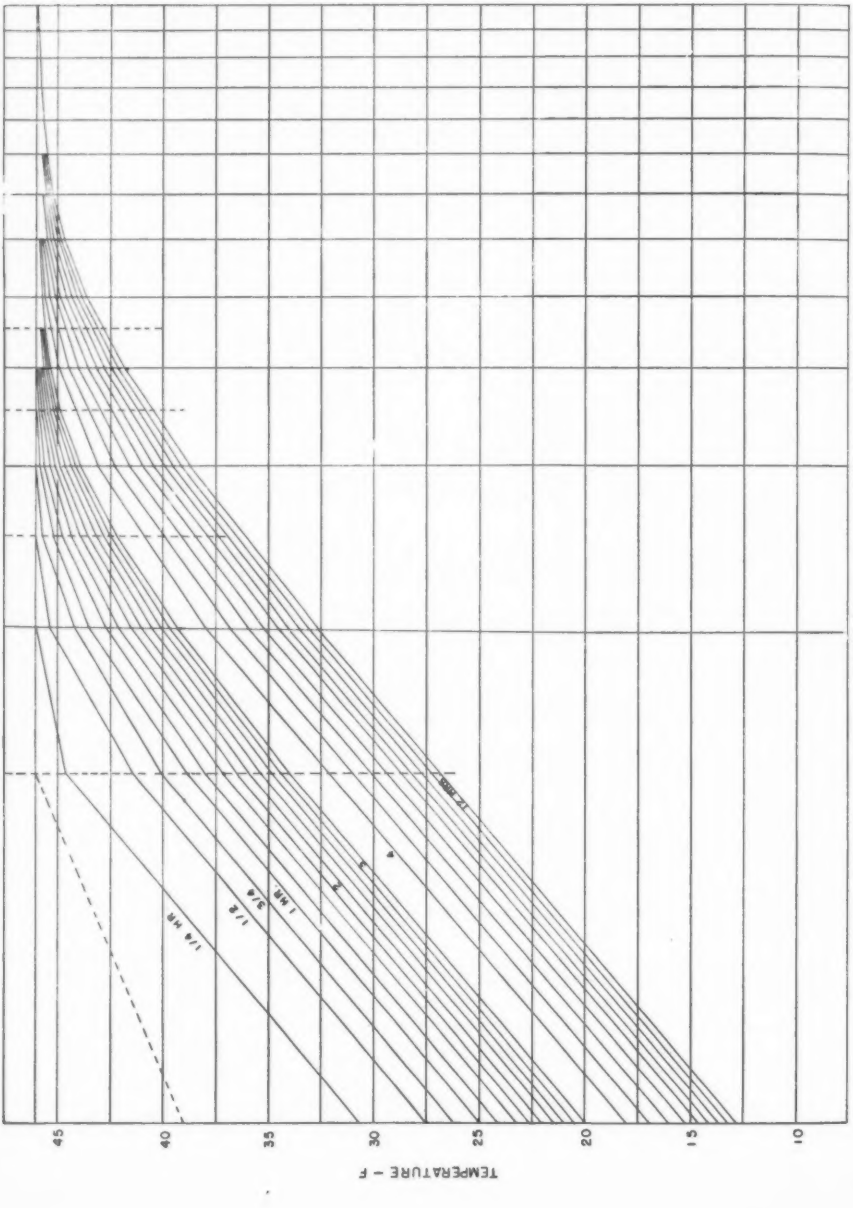

TEMPERATURE - F
45
40
35
30
25
20
15
10
1/4 HR.
1/2
3/4
1 HR.
2
3
4
12 HRS.
0.2364
0.4364
0.6364
0.8364
1.0364
2.0364
DISTANCE FROM CENTER OF TUBE - FT

TABLE 2—PERFORMANCE DATA, 1950-51

MONTH	RUNNING TIME HOURS	HEAT TAKEN FROM EARTH[a] BTU	HEAT DELIVERED[b] BTU	ELECTRIC ENERGY CONSUMED[c] Comp and Fan Kwhr	ELECTRIC ENERGY CONSUMED[c] Comp, Fan and Pump Kwhr	HEATING ENERGY RATIO	PRECIPITATION Normal In.	PRECIPITATION Actual In.	DEGREE-DAYS
September 1950..	7	292,000	429,000	40	43	2.92	3.1	6.5	28
October 1950....	29	1,270,000	1,837,000	165	180	2.99	2.8	1.6	75
November 1950..	140	6,100,000	8,570,000	750	824	3.04	3.8	5.7	735
December 1950..	122	5,320,000	7,560,000	655	719	3.08	3.7	2.5	994
January 1951....	130	5,680,000	8,180,000	729	797	3.00	4.2	10.5	843
February 1951...	91	3,970,000	5,550,000	470	518	3.16	3.7	4.2	688
March 1951.....	85	3,710,000	5,280,000	457	501	3.08	5.0	5.4	571
April 1951.....	61	2,660,000	3,790,000	330	362	3.07	4.4	4.0	351
May 1951.....	6	262,000	368,000	32	35	3.07	4.1	0.6	42
					Average	3.05		Total	4327

a Based on known operating time, pressures and manufacturer's data.
b Computed from heat absorbed from earth plus electrical input to compressor and fan motor.
c Assuming that the pump runs 50 percent in excess of compressor.

the cost of installation prohibitive. The results from the heating season reported indicate that, by use of a storage tank, the efficiency of an indirect expansion system is comparable to that of a direct expansion. This would not be the case in the absence of the storage tank. The thermal capacity of the storage tank and the intermittent operation of the compressor, made operation possible at a higher temperature in the earth heat exchanger than had been anticipated.

Test results indicate that the refrigerant circuit shown in Fig. 9 is satisfactory. Pressure drops were low, and conversion from one cycle to the other was performed without difficulty. The conversion could be more readily achieved however, by substitution of check valves for some of the globe valves shown. The experimental results reveal that the refrigeration equipment performed as predicted, and that all components of the system were well balanced. The capacity of the plant was adequate at all times, and the maximum running time during any day was 12 hr and occurred on February 2 when the outside air temperature dropped to a record of —22 F.

The cost of electrical energy for the heat pump performing with an average heating energy ratio of 3.05 is competitive with conventional systems in areas having low or medium electrical rates. The following table lists the approximate cost of electricity for which the energy expense of a heat pump is equivalent to that of heating systems utilizing conventional fuels:

GAS[a] AT	COAL[b] AT	OIL[c] AT	ELECTRIC RESISTANCE AT
$0.06 per 100 Ft³	$12.50 per ton	$0.15 per gal	$0.015 per kwhr
$0.009 per kwhr	$0.010 per kwhr	$0.017 per kwhr	$0.046 per kwhr

a Heating value, 1000 Btu per cu ft, efficiency 70 percent.
b Heating value, 12,500 Btu per lb, efficiency 50 percent.
c Heating value, 135,000 Btu per gal, efficiency 68 percent.

The maintenance cost of the heat pump in this installation has averaged approximately $8.00 annually for the past three years. The greatest portion of

this resulted from a loss of charge through a faulty compressor seal and is not expected to be recurring. The maintenance cost of a heat pump is expected to be comparable to that of an equal capacity refrigeration system operating a comparable number of hours. A local refrigeration service company was able to make all necessary adjustments and repairs. Consequently, the heat pump owner was not confronted with the increased expense of a specialized heat pump repairman. With proper mechanical maintenance, aging of equipment is not expected to alter the efficiency of a heat pump, therefore the average heating energy ratio reported during the test period is exepcted to prevail throughout the life of the plant. If copper had not been used for the heat exchangers, increased fluid flow resistance could have resulted from corrosion, thereby decreasing the plant efficiency.

The operating efficiency is dependent on the design of the refrigeration plant with excess pressure drop in any part of the system proving detrimental. The discharge pressure is controlled by the selection of the air conditioning coil and the quantity of circulated air, while the suction temperature is a function of the balance between the compressor and evaporator. The following tabulation serves to show the influence of evaporating temperature on the plant efficiency:

EVAPORATOR TEMPERATURE F	RATIO OF POWER SUPPLIED TO HEAT DELIVERED[a]
28	0.250
24	0.266
20	0.274
15	0.290
10	0.306

[a] Does not include the Joule equivalent of the heat generated by the motor.

The ratio of power required to the heat delivered at a suction temperature of 10 F increases approximately from 0.307 at 120 psig to 0.342 at 150 psig.

DESIGN OF EARTH HEAT EXCHANGER

As previously stated, the design of the earth heat exchanger represented the greatest single design problem. Excessive coil surface was to be avoided because of the high cost of copper tubing and the increased excavation expense. The first factor investigated was the depth of burial because increased depth assures the availability of a higher soil temperature but is counteracted by a greater excavation cost. This is true to a limiting depth, beyond which the summer recovery is less than the winter loss, thereby producing continuous cooling.[4] A depth of 6 ft seemed to be the optimum and also about the average reported in the literature. The yearly undisturbed temperature variation at this depth was calculated and found to be sufficiently high for a practical design. This computation was based on the assumption that the earth's surface temperature varied as a sine function passing through the maximum and minimum on July 15 and January 15, respectively. These values for the locality investigated were assumed to be 76 and 34 F, respectively, resulting in a mean surface temperature of 55 F. The temperature distribution at the depth in question was solved by the following equation from Jakob and Hawkins.[5]

TABLE 3—CALCULATED EARTH TEMPERATURES

MONTH	TEMPERATURE AT GROUND SURFACE F	UNDISTURBED TEMPERATURE 6 FT BELOW SURFACE F
April 15	55.0	48.2
May 15	65.5	52.4
June 15	73.2	56.4
July 15	76.0	61.5
August 15	73.2	64.0
September 15	65.5	64.1
October 15	55.0	61.8
November 15	44.5	57.6
December 15	36.8	53.6
January 15	34.0	48.5
February 15	36.8	46.0
March 15	44.5	45.9

$$\theta_{x,\tau} = \theta_o e^{-x\sqrt{\frac{\pi n}{\alpha}}} \sin\left(2\pi n\tau - x\sqrt{\frac{\pi n}{\alpha}}\right) \quad (1)$$

$\theta_{x,\tau}$ = the temperature excess (relative to the mean) at any time and depth, Fahrenheit degrees.
θ_o = temperature excess (relative to the mean) at the soil surface, 21 F deg for the particular problem, Fahrenheit degrees.
e = Naperian logarithm base, 2.718.
x = depth below the surface, feet.
n = frequency of the cycle, 1 per year or $1/(365 \times 24)$ per hour.
α = thermal diffusivity, square feet per hour.
τ = time measured from the point where the surface temperature reaches the mean while increasing and occurring on April 15 for this particular problem, hours.

The thermal conductivity and diffusivity of the soil were assumed to be 1.0 Btu per (hr) (sq ft) (F deg per ft) and 0.02 sq ft per hr, respectively. These values are possibly conservative since the soil is clay and is moistened by a normal annual precipitation of approximately 50 in.[4] The value of the diffusivity was later confirmed by Penrod and associates.[3] Table 3 presents the calculated monthly temperatures at the surface and at the 6 ft depth below the earth's surface.

The minimum undisturbed soil temperature is observed to be 45.9 F occurring on March 15. A minimum value of 45.6 F occurs between March 1 and 15, but this is not of sufficient importance to introduce fractional time intervals. However, as extreme weather is not likely after mid-February, 46 F may be considered as the minimum undisturbed soil temperature. This is a reasonably high value and consequently a burial depth greater than 6 ft is believed uneconomical.

Heat may be extracted from the soil by lowering the coil surface below 46 F with the magnitude being proportional to the temperature differential maintained. The minimum temperature of the coil is limited by the capacity of the compressor, and if the coil area is selected on the basis of a low surface temperature, a large displacement compressor is required. In general, the balance between the two is an economic problem and the best approach to the solution is the investigation of several possible combinations. The final com-

bination chosen for the Hopkinsville installation will be discussed as though it were a tentative one until the analysis is completed. By utilizing 480 linear ft of ⅞-in. O.D., type K copper tubing, the heat absorption rate per linear foot of tubing is readily calculated by dividing the total hourly heat absorption by the total coil length. This requires determination of the critical hourly heating demand. Three possible critical conditions must be investigated: (1) the design heat load which has a probable duration of only a few hours, (2) a rather heavy load over a prolonged cold period, and (3) the accumulated seasonal load. Each of these factors warrants an individual treatment.

The design heat load is defined as the continuous steady state rate of energy supplied to the building just adequate to replace the maximum anticipated rate of loss. It is general practice to express the design load on an hourly basis representing a daily average. The frequency of occurrence of the design load is usually from one to three times during a heating season, and the chance of two or more consecutive occurrences is indeed slight. Consequently, very rarely would the earth heat exchanger be required to supply the maximum load for a continuous period of more than one day.

The second critical condition occurs during a prolonged cold seige during which the average hourly load is considerably less than the design maximum but also much greater than the expected normal for the period. The duration of the moderately heavy load is not likely to exceed one month, but might exist at the time when the undisturbed earth temperature is a minimum.

The final critical load will produce the most serious effect at the time when the undisturbed soil temperature sags to its low point with extended time intervals being eliminated because of a decreasing demand for heat. The average load from the beginning of the season up to the time of the minimum undisturbed soil temperature is the significant load in producing the accumulated effect.

Three methods were used to analyze the critical loading conditions: (1) Ingersoll-Plass, (2) Binder-Schmidt, and (3) relaxation. It will be observed from the following discussion that one of the methods is more readily applicable to a particular analysis than another. The first critical loading condition was analyzed by all three methods with comparable results, while the others were investigated by the Ingersoll-Plass method only.

According to Ingersoll and Plass,[4,6] for a homogeneous soil at an initially uniform temperature t_o, the surface temperature t_s of an embedded tube of radius r ft is related to the rate of heat abstraction q Btu per (hr) (ft of length), the time lapse τ (hrs), and the physical properties of the soil by the equation,

$$t_s = t_o - \frac{q}{2\pi k} I(X) \quad \ldots\ldots\ldots\ldots (2)$$

$$X = \frac{r}{2\sqrt{\alpha\tau}}$$

$I(X)$ = Functional value of X. Evaluated in Reference 4.
k = thermal conductivity of soil, Btu per (square foot) (hour) (Fahrenheit degree per foot).
α = thermal diffusivity of soil, (square feet per hour).
τ = time, hours.
r = distance from tube center to point in question, feet.

The value of q for the first critical load is obtained from the calculated design heat loss of the building and the anticipated heating energy ratio of the system. Not all the energy needed to warm the building must be removed from the earth. The portion demanded of the earth q_e is obtained by multiplying the design heat load Q_d by $(R_{HE}\text{-}1)/R_{HE}$. (See Appendix A).

$$q_e = Q_d \left(\frac{R_{HE} - 1}{R_{HE}} \right) \quad \ldots \ldots \ldots \ldots \quad (3)$$

Based on an anticipated heating energy ratio of 3.00, and a calculated heat loss of 48,000 Btu per hr (70 F inside and — 5 F outside) the maximum heat sink for the 480 ft length of ⅞-in. O.D. coil becomes 66.7 Btu per (hr) (ft). Since this heavy load might occur when the undisturbed soil temperature is at its lowest, the value of t_o is taken as 46 F. Substituting these values, together with assumed ones for the physical properties of the soil into Equation 2, the surface temperature is found to be approximately 14 F, after a time lapse of 12 hr.

Fig. 12 shows the Binder-Schmidt[5,7,8] analysis for the same conditions with a resulting temperature at the tube surface of 12.7 F. This analysis is a continuous illustration of the temperature distribution throughout the soil in the proximity of the coil as time progresses. A very important observation is that the rate of surface temperature change is very rapid near the initial time but has decreased to a very low value after 12 hr. For this reason an extension of the investigation over a longer period was deemed unnecessary. The temperature gradient in the soil adjacent to the coil allows passage of the requisite heat, 66.7 Btu per (hr) (ft), and serves as a boundary condition defining the solution.

Fig. 13 illustrates a summary of the temperature distribution surrounding the coil by use of the relaxation method.[5,7] Just as for the case of the Binder-Schmidt analysis, the boundary at the coil surface is established by the temperature gradient required to drive heat into the coil at the necessary rate. Both methods would be applicable if the boundary were established by a known surface temperature, but if this were the restriction the rate of heat flow would not likely be constant. The relaxation method gives a surface temperature of 9 F after a time lapse of 12 hrs, a value somewhat lower than that given by the other methods, but comparable.

The second critical loading condition is best investigated by Equation 2. Application of either the Binder-Schmidt or the relaxation methods to this longer time period would require more work than computation by Equation 2. It is well to emphasize that only one point is located by each application of Equation 2 while analysis by the graphical methods simultaneously gives values for all points within the region under consideration. Therefore, if the temperature distribution were desired instead of only the surface temperature, the choice would probably not be Equation 2. The average heat load over the prolonged period was assumed as one half the design load, or 33.3 Btu per (hr) (ft) for this particular design. This load corresponds to 1125 degree-days per month, and it is doubtful that this value will be exceeded for the locality in question. Solving for t_s from Equation 2, the coil surface temperature is found to be 19.1 F after a period of one month, a value considerably higher than that for the 12 hr period at maximum load.

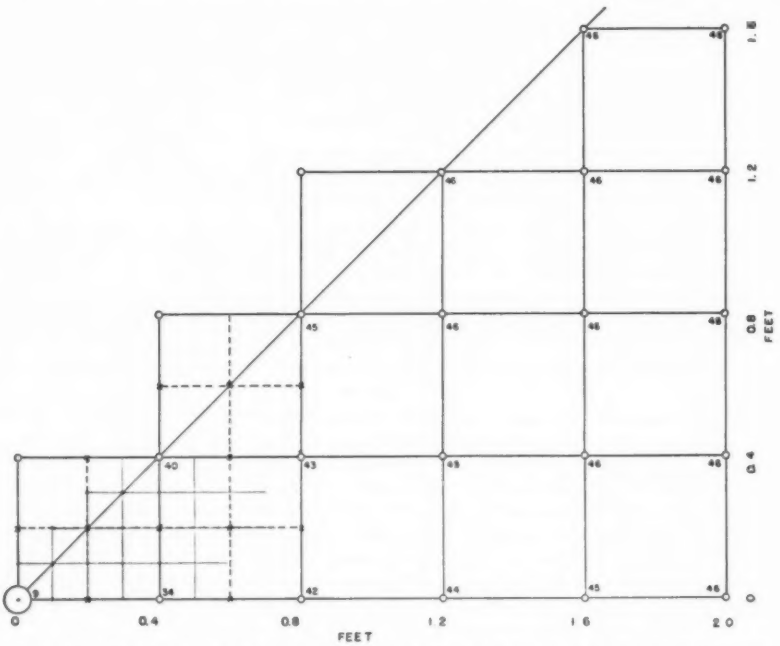


FIG. 13. TEMPERATURE DISTRIBUTION AROUND EARTH COIL AFTER 12 HR OPERATION BY THE RELAXATION METHOD

The third critical loading is investigated by Equation 2 with q being determined from the average hourly heat demand from the beginning of the season to the time that the undisturbed earth temperature reaches a minimum. The value was approximated as 18.4 Btu per (hr) (ft) by use of normal degree days. The corresponding coil surface temperature was calculated as 28.7 F, revealing that the short duration heavy load is the most severe. The possibility of superimposition of condition one and three must next be considered. If this superimposition should occur, the surface temperature might be expected to fall to approximately — 3 F, a value too low for practical design. This possibility was discounted on the basis of soil recovery data of Penrod and associates, (unpublished at the time of design, but included in References 1, 2 and 3). Later performance data justify the assumption, as shown by Table 1. By utilizing 960 ft of coil, the surface temperature after 12 hr operation at the design load (one half the original absorption rate per linear foot) would have dropped to 30 F rather than 14 F. The coil length of 480 ft was considered adequate.

CONCLUSIONS

This residential earth heat pump installation operated successfully during the 1950-51 heating season with the outside air temperature falling as low as — 22 F.

The operating cost of an earth heat pump with a heating energy ratio of 3.05 is competitive with conventional systems in areas having low or medium electrical rates.

The Binder-Schmidt, Ingersoll and Plass, and relaxation methods may successfully be employed to design an earth heat exchanger for a heat pump system.

ACKNOWLEDGMENTS

The author wishes to thank Dean D. V. Terrell, College of Engineering, University of Kentucky and Al Wettstein, superintendent, Hopkinsville Electric Plant Board, for making available instruments for the tests. The interest and suggestions of Prof. E. B. Penrod, Head, Department of Mechanical Engineering, University of Kentucky, were greatly appreciated. The aid of the R. A. Wilson Co., Russellville, Ky., was invaluable in obtaining the refrigeration equipment. Appreciations are extended to Jesse F. Baker, owner, for patiently and interestingly recording data.

REFERENCES

1. Earth Heat Pump Research, Part I, by E. B. Penrod, O. W. Gard, C. D. Jones, H. E. Collier, and R. N. Patey (*Engineering Experiment Station Bulletin,* University of Kentucky, December 1949, pp. 1-63).

2. Performance of an Earth Heat Pump on Intermittent Operation, by E. B. Penrod, E. L. Dunning, and H. H. Thompson (*Kentucky Academy of Science Transactions,* Vol. 13, No. 2, October 1950, pp. 83-99).

3. Earth Heat Pump Research, Part II, by E. B. Penrod, R. B. Knight, and Merl Baker (*Engineering Experiment Station Bulletin,* University of Kentucky, December 1950, pp. 1-103).

4. Theory of the Ground Pipe Heat Source for the Heat Pump, by L. R. Ingersoll and H. J. Plass (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 339).

5. *Elements of Heat Transfer and Insulation,* by M. Jakob and G. A. Hawkins (John Wiley & Sons, Inc., New York, 1950).

6. Theory of Earth Heat Exchangers for the Heat Pump, by L. R. Ingersoll, F. T. Adler, H. J. Plass, and A. C. Ingersoll (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 167).

7. *Heat Transfer,* Vol. I, by M. Jakob (John Wiley & Sons, Inc., New York, 1949).

8. *Refrigeration and Air Conditioning,* by B. F. Raber and F. W. Hutchinson (John Wiley & Sons, Inc., New York, 1946).

APPENDIX A

Development of Equation 3

By definition the heating energy ratio is

$$R_{HE} = (q_e + W)/W \quad \text{(A-1)}$$

where

q_e = heat energy taken from the earth.
W = electrical energy supplied to the motors.

By rearranging Equation A-1,

$$W(R_{HE}) = (q_e + W) \quad \text{(A-2)}$$

or

$$q_e = W(R_{HE} - 1)$$

or

$$q_e/(W + q_e) = W(R_{HE} - 1)/(W + q_e) \quad \text{(A-3)}$$

Substituting $1/R_{HE}$ for $W/(W + q_e)$,

$$q_e = (q_e + W)\,[(R_{HE} - 1)/R_{HE}] \quad \text{(A-4)}$$

Equation A-4 becomes Equation 3 of the paper by substituting Q_d for $(q_e + W)$

APPENDIX B

Calculations

Undisturbed Soil Temperatures

Given the following conditions:

$\theta_o = 21$ F.
$x = 6$ ft.
$\alpha = 0.02$ sq ft per hr.

1 cycle per year = 1/8760 cycles per hr = 1/12 cycle per month.

Find undisturbed soil temperature $t_{x\tau}$ on May 15.

$$\sqrt{\frac{\pi n}{\alpha}} = \sqrt{\frac{\pi \frac{1}{8760}}{0.02}} = 0.134 \text{ ft}$$

$$\theta_o e^{-x\sqrt{\frac{\pi n}{\alpha}}} = 9.4$$

Values of τ for various months are as follows: May 15 = 1, June 15 = 2, July 15 = 3, and April 15 = 12.

Substituting in Equation 1 of the paper:

$\theta_{x_\tau} = 9.4 \sin[(\pi/6) = 0.804] = -2.6$ F deg.
$t_{x_\tau} = 55.0 - 2.6 = 52.4$ F.

Coil Surface Temperatures by the Ingersoll and Plass Method

For Critical Loading Condition 1

The various terms are as follows:

$\tau = 12$ hr.
$r = 7/8/2{:}12$.
$q = 66.7$ Btu per (hr) (ft).
$k = 1$ Btu per (hr) (sq ft) (F deg per ft).
$\alpha = 0.02$ sq ft per hr.

Then

$$X = \frac{0.0364}{2\sqrt{0.02 \times 12}} = 0.0372$$

$I(X) = 3.012$ (Page 340, Reference 4).

Substituting the proper values in Equation 2 of the paper:

$$t_s = 46.0 - \frac{66.7}{2\pi\,(1)}\,(3.012) = 46.0 - 32.0 = 14.0 \text{ F}$$

For Critical Loading Condition 2

The various terms are as follows:

τ = 720 hr.
q = 33.3 Btu per (hr) (ft).

Then

$$X = \frac{0.0364}{2\sqrt{0.02 \times 720}} = 0.0048$$

$I(X) = 5.06$ (Page 340, Reference 4).

The coil surface temperature, by substituting proper values in Equation 2, is,

$$t_s = 46.0 - \frac{33.3}{6.28\,(1)}\,(5.06) = 46.0 - 26.9 = 19.1 \text{ F}$$

For Critical Loading Condition 3

τ = 3410 hr.
q = 18.4 Btu per (hr) (ft) (Average between Sept. 21 and Feb. 15).

Then by substituting proper values in Equation 2, the coil surface temperature is,

$$t_s = 46.0 - \frac{18.4}{2\pi\,(1)}\,(5.84) = 28.8 \text{ F}.$$

DISCUSSION

J. DONALD KROEKER, Portland, Ore.: This heat pump installation has a number of very interesting features. In my experience the combination of a storage tank with a ground coil installation is novel. It does have a number of advantages, the chief one being the accessibility of refrigerant piping so that, if ice develops, it can be discovered. If leaks occur, which have been a problem in ground coils, they are isolated and accessible for repair. The leakage of water into the ground in the event of the corrosion of the piping in the ground is of less consequence and certainly holds fewer problems from the standpoint of continuous operation through the winter after the leaks develop.

Here is an installation that has been, in my opinion, over-designed on the basis of factors for transmission of heat in the ground which however, offers advantages. I see a declination of that admission in the paper, but the apparent proof of it is the fact that it was not necessary to use anti-freeze solution for circulation in the ground coils. Incidentally, the length of 480 ft of pipe used in this installation is considerably less than that found by trial and error—mostly error—in the Pacific Northwest as being suitable for an installation of this size.

The heating energy ratio disclosed is quite high for the average seasonal ratio on ground coil operation as we know from experience. It is noted that reliance has been placed on data development on a basis for prediction of ground temperatures. I feel it is necessary for us to gain information on actual ground temperatures and actual conditions on installations such as this which are actual installations in use. I hope the authors will extend their studies in this direction.

I should like to have information on the installation cost involved. It is stated that the installation cost was not charged entirely to heating. The over-all cost figure would be of interest in approaching a problem of this kind.

G. O. G. LÖF, Denver, Colo.: (1) What type of soil was encountered in the ground surrounding the coil? (2) Was the circulating pump controlled separately from the compressor? If so, what was the primary control element for the circulating pump?

S. S. SANFORD, Detroit, Mich.: What was the horsepower and what was the energy consumption of the water circulating pump? The part of Table 2 giving electric energy is not clear. The column labeled *Fan kwhr* appears to include the compressor. On page 383 there is a statement, *The results from the heating system reported indicate that, by use of a storage tank, the efficiency of an indirect expansion system is comparable to that of a direct expansion.* Does that statement relate only to the temperature level at which the system operates, or does it also take into account the electric consumption for the water circulating pump?

There is also a statement that the capacity of the plant was adequate at all times, and that on the day when the temperature dropped to −22 F the plant operated only 12 hr. Does this imply that a smaller ground coil could have given adequate capacity but with a somewhat lower heating energy ratio and a lower seasonal performance factor? Is there any record of the performance factor or heating energy ratio during the two following seasons, 1951-52 and 1952-53?

Lastly, why was the tank made so large that the use of an anti-freeze solution was found unnecessary?

W. F. FRIEND, New York, N. Y.: I have three questions. (1) Why did the air circulating fan require only ¼ hp, when the low air temperatures (around 105 F) and the large volumes handled are considered? The fan power in such cases usually is considerably higher.

(2) With this indirect expansion system involving the circulation of water through the ground coils, it should have been possible to measure the heat delivered to the dwelling by metering of water flow and taking the temperature difference in and out of the ground. Instead, the heat quantities were determined from manufacturers' characteristic performance curves for the heat pump. What was the basis for choice of method?

(3) What might have been the effect on system performance of increasing the ground coil 50 percent or decreasing the ground-surface area or the lineal footage of pipe by an equal percentage?

G. W. HUMMEL, Phoenix, Ariz.: I would like to ask the author what the over-all cost of the operation was per day.

R. C. CHEWNING, Portland, Ore.: I would like to ask if any consideration was given to the use of a standard type water chiller from which water would be circulated to the storage tank. The temperature difference between the evaporating refrigerant and the leaving water of the chiller could be held within 8 deg rather than the 19 deg indicated by the minimum water and suction temperatures of 36 F and 17 F respectively in the system installed. An increase of 10 F in the operating suction temperature would increase, significantly, both the efficiency and the capacity of the unit.

M. K. FAHNESTOCK, Urbana, Ill.: The only point I would like to make is in regard to comfort air-conditioning as a year 'round proposition. It is becoming more and more popular for residences and this paper should have been extended to include information and data on the cooling cycle, in addition to the heating cycle. In our experience with an air-to-air heat source heat pump in residential work we were always interested in year 'round performance and I believe such interest is general.

While these comments do not specifically pertain to the data presented, I would like to see the work continued and data obtained or published on the conditions produced and maintained inside of the residence itself, during both the heating and cooling cycles. This paper merely states that the conditions inside of the residence were satisfactory.

We know there are very definite problems in applying heat pumps to residences. One is the relatively large volumes of air which have to be handled. There is also the problem of the difference in the air volumes required for heating, and those

required for cooling. When we have a test set-up like the one described in the paper, we should take advantage of it and obtain complete data for both the heating and cooling seasons.

AUTHORS' CLOSURE (Y. S. Touloukian, West Lafayette, Ind.):* Mr. Kroeker's question concerned overdesign; my personal feeling is that the unit is not overdesigned as far as the evaporator coil is concerned which is properly sized to supply the energy for which designed. However I think the heat load on the house has been overestimated. The unit was designed for a much larger house.

This was a custom-made job, and the author did not meet all expenses. This will be apparent from the acknowledgment.

The coating of ice that was mentioned was not on the buried coil, but rather, on the coil submerged in the ballast tank. The depth of the burial and the distance between coils were as stated in the paper.

As to soil characteristics, the soil is essentially a clay as indicated by the author. The values of the thermal properties were as follows: The thermal conductivity and diffusivity were assumed to be 1 Btu per (hr) (sq ft) (F deg per ft) and 0.02 sq ft per hr respectively. Precipitation is about 50 in. per year. The control of the circulating pump was independent from that of the compressor. The pump ran somewhat longer than the compressor.

As to the suggested use of a water chiller to increase further the efficiency of the system, any scheme that will enable the water coil temperature to run at a high value would, of course, be better for the compressor. In the consideration of favorable electric rates, my personal experience has indicated that 1½ cents per kw hr should be taken as a maximum before you can start competing favorably with conventional fuels. I imagine that Hopkinsville, Ky., has such rates or lower.

I do not know whether data since 1951 have been compiled by the occupants of the house. If such data are available they should be published not only for the heating season but also for the cooling season.

I agree that the energy abstracted from the soil could have been given, as the author had thermometers at the inlet as well as at the outlet of the coils. There should be no problem in metering the water flow in the ground coil and, thus, obtain actual calorimetric data on energy absorbed from the coil. This has not been reported, and I have had no opportunity to obtain further details from the author.

What would happen by increasing the ground coil length by 50 percent? The author comments on this possibility in the paper immediately preceding the conclusions. You will remember that the coil used was 480 ft long. By utilizing 960 ft of coil the surface temperature after 12 hr operation at the design load would have dropped to 30 F rather than to 14 F. That means the higher evaporation temperature would be indicated here.

I might make some personal comments concerning ground *vs.* air-to-air applications. It was of interest to me to note a year or so ago that a nation-wide survey showed that almost without exception universities were working with the ground source heat pumps, while industry was working with air-to-air systems.

I am not going to draw any conclusions from these findings. I personally prefer ground applications. Air-to-air applications are geographically limited. Ground applications are more universal in character, as the temperature gradient in the ground does not vary appreciably across the United States at a depth of about 8 ft from the soil surface. Any design procedure and physical constants developed will be almost universally applicable. There are some geographical considerations with the ground; areas of higher precipitation would offer more favorable locations than those of lower precipitation, but the differences will not be nearly as great as in air temperatures. Another possibility with ground application is the storage of energy

* These closing remarks were made by Dr. Touloukian following his presentation of the paper and are supplemented by Dr. Baker's closure on p. 394.

for short term utilization. Although a more complex medium than air, the ground is a more versatile one from the standpoint of heat pump application. The physical constants are not well known nor can they be assumed as constant once they are determined.

The author made a brief comment about the importance of luck in his design. I would like to underscore that. At Purdue University we have been making theoretical heat pump analyses for several years. We did not have the funds to make installations and conduct expensive experimentation. I believe a sound and rational analysis of idealized systems, that takes account of the imperfections of equipment is of vital importance. There is a good prospect that design manuals for ground equipment well within the safety factors that are presently used in engineering practice will be available in the near future. I hope in the next five years or so the engineer will have available design procedures of considerable merit.

Soil constants will have to be determined, and agreement in procedures must be developed for uniformity in establishing soil designations. There is a considerable amount of data on temperature gradients in undisturbed soils from the surface to a depth of 20 ft. All these data show a surprising similarity. We need more information on what happens if the ground is disturbed. What about ground temperature distributions under buildings where considerable differences can be expected?

We also have some idea as to the recovery times for double or triple coils operated intermittently. This involves the operation of one set of coils, a change after a certain period to coil number two, and finally to coil number three. Such a system permits recovery where conditions are not very conducive to rapid recovery of a single coil. This seems to show considerable promise. However, the economics of three coils against one is the deciding factor as to the practicability of such a system.

AUTHOR'S CLOSURE (Merl Baker): I wish to thank Dr. Touloukian for presenting this paper and offering the excellent concluding remarks. I will attempt to reply to some of the questions introduced, the answers to which Dr. Touloukian could not have known.

The time delay control governing the ⅓ hp circulating pump was controlled by a thermal element located on the discharge line of the compressor. The delay was almost constant at 10 min and accordingly the total running time of the pump was dependent on the cycling frequency of the compressor. The pump started within one minute after the compressor.

Increasing the size of the water storage tank will continue to improve the performance of the heat pump, however, a practical limit is ultimately reached. The tank chosen was limited in size by the available space for location. It is doubtful that a smaller tank should be considered for an installation of this size.

The electrical rate was less than one cent per kw hr and the average operating cost per day was approximately $0.25.

Both the energy consumption of the fan and pump motor was included in the performance ratio. The energy to the pump was not metered but determined by an instantaneous measurement multiplied by the operating time. It was assumed that the pump ran 50 percent longer than the compressor.

Professor Fahnestock's remarks are very appropriate. In our central and northern latitudes, the residential sensible cooling load is usually about one-third of the heating load. For heating, air is taken from the floor at 65 F and delivered at 105 F, giving a differential of 40 deg.

For summer operation, the cooling capacity of the plant would be sufficient to lower the temperature of the same air quantity about 30 deg. Good air distribution is definitely a problem in utilizing the excess summer capacity of a heat pump. The ideal heat pump system would consist of two or more compressor units which would offer load modulation for both winter and summer, however, this design would increase the first cost.

1484

No. 1484

MOISTURE MOVEMENT IN SOILS DUE TO TEMPERATURE DIFFERENCE

By W. A. HADLEY* AND RAYMOND EISENSTADT**, NEW YORK, N. Y.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Columbia University, New York, N. Y.

FOR THE PAST three years a series of experiments have been carried on at Columbia University to determine how and why moisture moves in granular media such as soils under a temperature difference. This problem became of interest during a study of heat pumps that used evaporators buried in the ground. These so-called *ground coils* performed in a manner that could not be explained by existing theory. For one thing, the coils seemed to be able to absorb more heat than expected from calculation by accepted heat transfer theory. In fact, the heat absorbed under certain conditions might be 50 percent greater than predicted. Therefore, it seemed that either the theory or the experimental figures on the heat pumps were wrong, or something was happening that had not yet been taken into account.

It was also observed that the ground coils seemed to attract water. When one of them was dug up after a winter of operation the soil around it looked like glass and had to be chipped off. When this soil thawed there was so much water in it that the sample showed saturated mud with a layer of clear water standing above it. In other installations the ground heaved up over the ground coils enough to warp and throw out of plumb the buildings being heated. It was thought that the movement of water to the ground coil and the excess heat reaching it were probably factors of the same problem. A search of the literature revealed that W. O. Smith[1] and G. J. Bouyoucos[2] had shown that a temperature difference in a soil caused water to move toward the lower temperature but how it moved, why it moved, and how much water moved was still in doubt. It was decided to try to find answers to some or all of these problems in the proposed studies at Columbia.

One difficulty in conducting research with soils is that soils are not uniform and have particles of such irregular shapes and sizes that calculations of results

* Associate Professor, Department of Mechanical Engineering, Columbia University.

** Graduate Student, Department of Mechanical Engineering, Columbia University.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

[1] Thermal Transfer of Moisture in Soils, by W. O. Smith (*Transactions of the American Geophysical Union*, 1943).

[2] G. J. Bouyoucos (*Michigan Agricultural Experiment Station, Tech. Bull. 22*, 1915, pp. 1-63).

are difficult. To avoid this difficulty small perfectly spherical glass beads of 0.01 in. diameter resembling white beach sand were used. The particles, being spherical, regular in size, and uniform in other physical properties, permitted calculations to be made and results to be reproduced.

Since all tests were made in this idealized soil medium all references to the performance of actual soils might be in error. It is believed that sands might show a behavior very similar to that obtained with the glass beads while clays might show considerable differences. All references to soils in this report there-

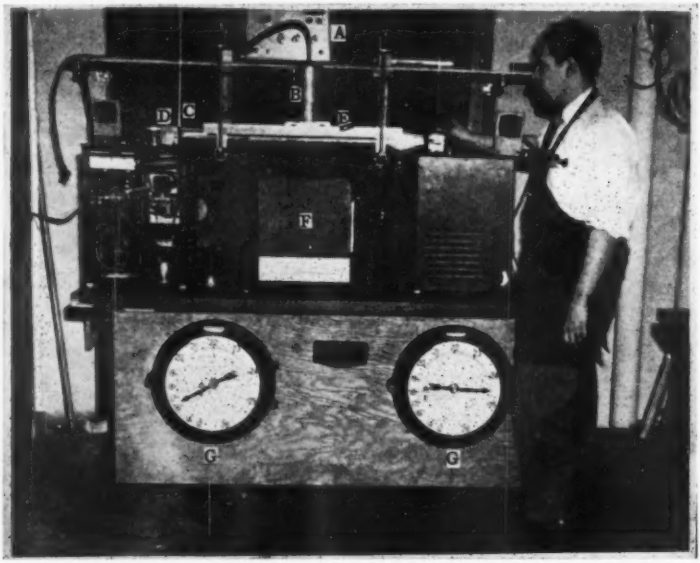


FIG. 1. ARRANGEMENT OF TEST APPARATUS

A. Geiger counter
B. Lead shielded Geiger tube
C. Specimen tube and insulation
D. Brass blocks in oil bath
E. Steel rails and scale
F. Thermocouple cold junction box
G. Temperature controller

fore apply to the material tested, since natural soils were not tested and compared with the ideal soil.

In order to determine how the water moved, a radioactive tracer technique was developed. A very small quantity of radioactive cobalt 60 (the material now being used to treat cancer) in the form of cobaltous nitrate added to water (0.1 gm per liter) would have no appreciable effect on the properties of the water but could be detected by a Geiger counter located outside of the thermally insulated specimen pipe containing the soil. This technique had two main advantages:

1. It could be determined whether there was a uniform moisture distribution in the soil specimen at the beginning of a test without taking samples, and weighing, drying, and reweighing. Drying and weighing destroys the soil specimen for test

purposes. With the method used in these tests it is only necessary to pass the Geiger tube over the radioactive soil model to determine the initial moisture distribution. The specimen can then still be used for further tests.

2. The form in which the moisture moves can be determined. If all the moisture moves in the liquid form, the moisture content at the end of the test and the radioactivity will be proportional, because the radioactive salt will be carried by the liquid. If all the moisture moves in the vapor phase, the salt will be left behind by the vapor and there will be no change in radioactivity during a test, but, at the end of the test, weighing and drying samples will show that there has been a moisture movement. If part of the moisture moves as a liquid and part as a vapor, a condition will be found half way between the above. Incidentally, by this method, liquid movements can be observed at all times during a test.

The test apparatus is shown in Fig. 1. The boxes shown at either end of the table in Fig. 1 each contain heating coils and refrigeration equipment so that a temperature drop can be produced and maintained along a plastic pipe that connects them. This 1½-in. internal diameter, ⅛-in. thick plastic pipe is well insulated on the outside and is filled with the previously described glass beads used as the soil specimen. Thermocouples are inserted along the length of the pipe to measure the temperature of the soil, and provision is made in the pipe so that samples may be taken rapidly at the end of the run to determine the final moisture content by weighing and drying. The Geiger tube and counter are at the top center of the picture, Fig. 1.

Space does not permit inclusion of all of the results attained during the three year study. However, some of the conclusions should be of interest.

The most important conclusion reached was that there is a critical moisture content, hitherto unknown, that separates two distinct methods of moisture movement.

In the soils tested this moisture content was about 4 percent moisture by weight of the total dry weight. The moisture movement in soils wetter than this was totally different than the moisture movement in dryer soils. In general, most soils are wetter than the critical, but dryer soils are found.

Fig. 2 is a good example of these so-called wet soils while Fig. 3 is an example of the behavior of the dry soils (the abbreviation C/M is counts per minute). If Fig. 2 is studied, certain conditions become apparent in light of the previous discussion.

1. The moisture movement at the cold end is apparently all in the liquid phase because the curve of radioactivity coincides with the curve of final moisture content. (There is some vapor movement at the hot end, a separate phenomenon that is being studied in greater detail as it is of considerable interest in underground electrical cable work.)

2. The increase in moisture content coincides roughly with the ice point, or 32 F temperature, in the pipe. In other runs where temperatures were maintained above 32 F no moisture movement in the wet samples was observed.

3. It will be noted that the moisture content of the whole curve is uniformly depressed except in the region adjacent to the freezing where a sharp depression occurs. It appears that the moisture build-up at the cold end is obtained initially from the adjacent unfrozen zone which in turn is replenished from the rest of the pipe.

In contrast to conditions in Fig. 2, Fig. 3 illustrates the effect of dry soil (*i.e.*, soil having less than critical moisture content). It will be noted that: (1) all of the moisture movement is apparently in the vapor phase, because there is no change in radioactivity from the beginning to the end of the run while

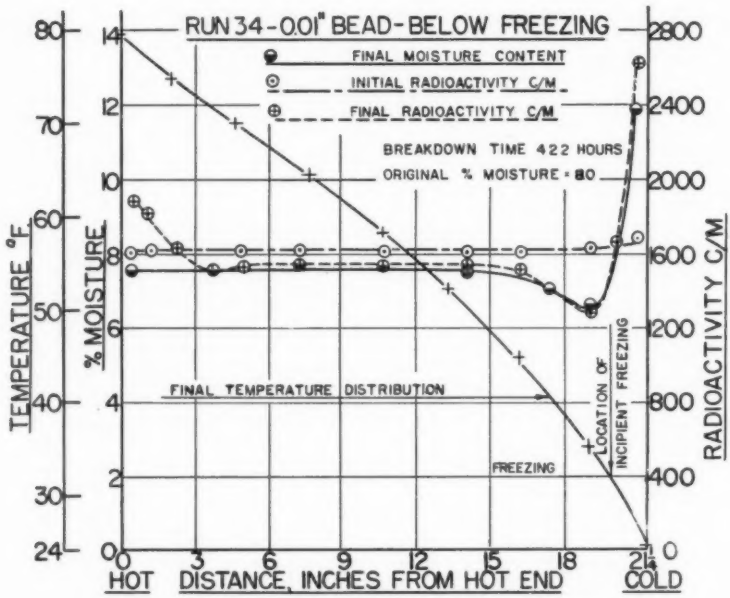


FIG. 2. TEMPERATURE, MOISTURE, AND RADIOACTIVITY VS. DISTANCE IN WET SOIL

there is a change in moisture content; and (2) this is not a phenomenon associated with freezing. The conditions described, although previously suspected in part, were until now unverified and in some cases unexplored.

These two phenomena and the apparent critical moisture content condition that separates them are being studied further with the expectation that the mechanism involved will be the subject of another paper in the near future.

The information uncovered by this investigation is of threefold interest to heating and ventilating engineers. This work bears a direct relationship to *heat pump ground coils*, to *air conditioning in mines*, and to an understanding of that *Arctic soil condition* called *permafrost*. How much direct quantitative help can be derived is still in question, because it should be remembered that these initial laboratory tests by necessity are oversimplified as compared with conditions obtained in undisturbed natural soils. However, additional studies can be made that will provide information that can be used directly in calculations. Bearing this in mind, the following significant facts may be noted:

With respect to heat pump ground coils, it is possible (and even probable) that a considerable amount of heat is carried directly to the ground coil by the water movement. In the laboratory apparatus it was calculated that the heat transferred by conventional heat transfer amounted to 10 Btu per (sq ft) (hr) in the wet soils while the water migrated in the wet soils at an average rate as high as 15 grams per (sq ft) (hr) yielding a latent heat of fusion of the order of 5 Btu per (sq ft) (hr).

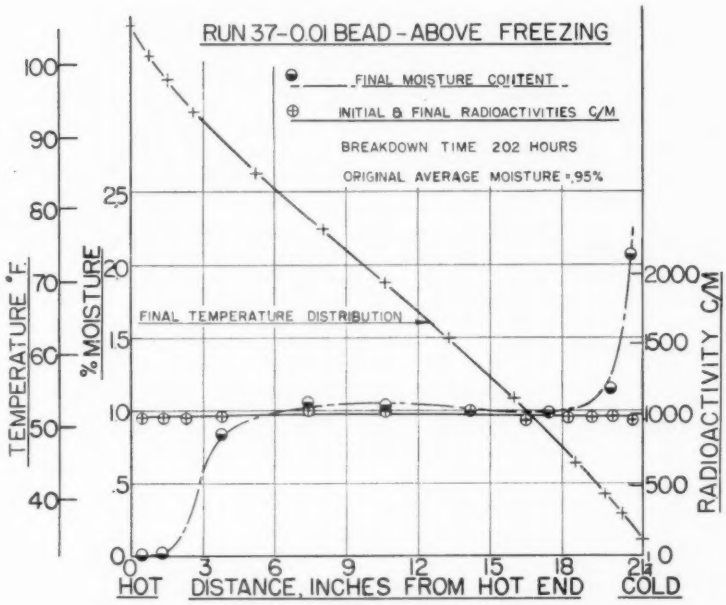


FIG. 3. TEMPERATURE, MOISTURE, AND RADIOACTIVITY VS. DISTANCE IN DRY SOIL

If this condition also exists in the earth, the discrepancy between theory and practice is accounted for. Conventionally, heat pumps are operated in an off-on cycling fashion. Fig. 4 shows the effect of starting and stopping. It should be noted that this method of operation seems to accentuate the phenomena and that in the *off* period a considerable amount of moisture appears to melt and is left in a position easily available for another moisture migration, thus increasing the amount of moisture adjacent to the cold face. In fact, at 112 hr (in Fig. 4) the rate of increase of the moisture at the cold end was very slow, indicating that an equilibrium point was not far away. At that time the cold end temperature was allowed to rise to room temperature and the resulting curve is shown at 142 hr. At 142 hr the temperature at the cold end was again reduced. This caused the curve to rise far above the level the original curve had been approaching. Therefore, *on-off* operation should give considerably more heat than is calculated on the basis of the amount of moisture found frozen around the ground coil at the end of a heating season.

In dry soils (which in this idealized case means soils below about 4 percent moisture) the total amount of heat transferred by the moisture is considerably less, even though the vapor furnishes both its latent heat of evaporation and its latent heat of fusion. The moisture movement in this case can account for about 0.8 Btu per (sq ft) (hr) or about 16 percent of the amount transferred in wet soils. However, the total amount of heat transferred in dry soils is also a smaller portion of that in wet soils due to the smaller thermal conductivity

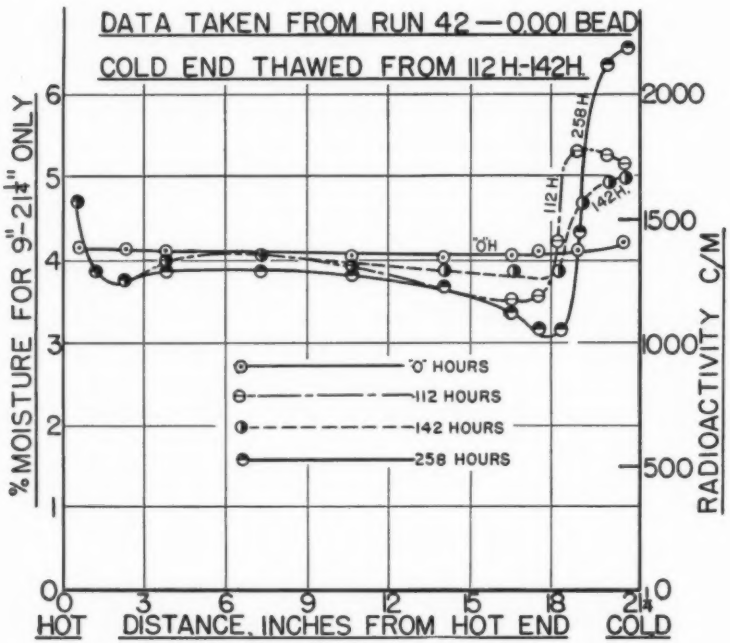


FIG. 4. RADIOACTIVITY VS. DISTANCE, FREEZING—THAWING RUN

of dry soil. Hence, the percent of the total heat that might be chargeable to moisture movement may still be high. However, the total heat transferred in any case is so small that the installation of heat pump ground coils in soils below the critical moisture content is therefore doubly undesirable. Determination of the value of the critical moisture content for various types of real soils would be a valuable project.

An air conditioned mine shaft resembles a heat pump ground coil in that it is a cold conduit in the earth. There has been some speculation as to the amount of moisture that might migrate into mine shafts due to the temperature gradient. On the basis of the present study this consideration can be largely eliminated because no appreciable moisture movement was found in the wet soils without freezing temperatures and the soils will rarely, if ever, be below the critical moisture content.

Permafrost is the term applied to the portion of the ground in the Arctic that stays frozen all year round. It creates a difficulty when a heated building is constructed in the Arctic as the heat from the building melts the permafrost on which the building foundation rests. This study gives some insight into the nature of permafrost. The frozen portion of the soil in Fig. 2 and the part remaining frozen in Fig. 4 are analogous to permafrost. This would indicate that high moisture contents might characterize permafrost in soils above the critical while, if soils could be maintained below the critical no problem might

exist. These statements are at best exploratory on the basis of the present work. However, additional work in this field on permafrost might be not only pertinent but highly productive.

DISCUSSION

N. B. HUTCHEON, Ottawa, Ont., Canada (WRITTEN): The work reported in this paper represents still another attack on the very difficult problem of moisture movement materials. The radioactive method employed is new and currently popular, but is not necessarily, because of this, more revealing or more useful than other approaches which have been made.

It is not clear from the paper whether the authors are familiar with the very great amount of work already done in this field. It may be supposed that such references were omitted in the interests of brevity, but the conclusions would appear to have been reached without regard for some of the more important contributions already presented in the literature.

Certainly nothing presented in the paper can be considered sufficient evidence on which to base any generalized conclusions at this time, nor has the method been demonstrated as sound. The complete results, if made available, would undoubtedly be interesting, and extremely valuable in certain directions, but the phenomena under study have already been shown to be so complex that it is very doubtful if any far-reaching conclusions ought to be drawn from any one series of experiments carried out in one particular way.

It must first of all be noted that the radioactive tracer is the cobalt ion, and not the water. It is already well recognized that cations can be involved in surface adsorption and in ion exchange and that they can diffuse. No recognition is given in the paper to these possibilities which, if they exist, would at once invalidate the basic assumption that the cobalt ion remains responsive only to the liquid water phase and to its movement. Gurr, Marshall and Hutton—in a paper in *Soil Science*, November 1952, on "Movement of Water in Soil Due to a Temperature Gradient"—recognized the difficulties with cations and used the chloride ion as a tracer, on the same fundamental basis as the authors. Gurr, Marshall and Hutton were able to get results by direct chemical analysis for chloride ion, without the complications of the radioactive tracer technique, or of the possible cation exchange and cation diffusion which complicate the interpretation of the results. Their paper also provides a very useful list of references to many of the important sources of information in the soil science field.

The results obtained by Gurr, Marshall and Hutton indicate that, except for very dry and very wet soils, the chloride ion concentrated at the hot end, indicating a vapor movement to the cold end, accompanied by condensation and liquid return to the hot end carrying the salt. The tests by the present authors indicate increased radioactivity at the cold end, presumably as a result of liquid movement to the cold end with freezing conditions. According to the authors there is also a sharp separation at about 4 percent moisture content in the glass beads used, between liquid and vapor mechanisms of moisture movement.

A mechanism similar to that found by Gurr, Marshall and Hutton was postulated by Paxton and Hutcheon on the basis of studies of moisture migration in wet sawdust, as reported before the A.S.H.V.E. in 1952.*

In view of these and of other results which might be quoted, it would seem to be wise to reserve judgment at present as to the existence of a sharp separation between vapor and liquid movement, the moisture contents in various soils at which maximum moisture shifts can be produced, and the nature of the mechanisms operating at various moisture contents in different soils.

---

* Moisture Migration in a Guarded Hot Plate, by N. B. Hutcheon and J. A. Paxton (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 301).

The authors' comments in the paper about permafrost are more optimistic than helpful, since one of the things which is characteristic of permanently frozen ground is the fact that drainage of liquid is hampered by the permanently frozen soil beneath the active layer. All precipitation tends to be trapped and to collect in the surface layers. Thermally actuated moisture movement may be a factor in permafrost but the two basic factors are low average temperature and precipitation.

Y. S. TOULOUKIAN, Lafayette, Ind.: The use of radioactive tracers as a tool in studying diffusion problems is a very promising one and it is most heartening that an engineering effort has been made in this direction. I am somewhat disturbed not to find many other studies in this field in the references. I hope that the authors are well informed on other activities going on using radioactive tracers as well as analytical and experimental determinations of moisture diffusion in soils in an attempt to determine the mechanism as well as to secure experimental data to work back and develop the theory from it.

The one thing that bothers me somewhat is whether we should work with disturbed or undisturbed soils. There is considerable evidence, of sufficient reliability, that disturbed and undisturbed soils are not the same thing; thermal conductivity for instance varies by as much as 75 percent under identical thermal and moisture conditions and densities depending on whether the soil is disturbed or undisturbed. There is a question whether disturbed soil ever becomes undisturbed again.

These are problems of considerable interest, because in ground installations a ditch must be dug and back filled. Such a ditch must be considerably wider than the pipe it is to receive. This leaves a critical region around the coil where the soil is disturbed. Outside of this region, of course, the soil is undisturbed. To date most of the determinations of physical constants have been on disturbed soils.

As to the true mechanism of vapor diffusion I do not have the same optimistic view as the authors have in trying to arrive at the answer. Many others, as he indicated, have attempted with varying degrees of success to arrive at an answer. Most probably the true story lies in none of the existing theories as a whole but somewhere in between. The problem is very difficult to investigate experimentally. The idealized situation is tolerable from an academic approach. However data obtained from idealized systems are hardly adequate for use in design. Eventually we have to come to a standardization of soil compositions and assume average values which might be within 20 percent.

How vapor migrates, whether it migrates solely as a liquid or a vapor, I do not think is a proper question. If nothing else be considered we know that at any one time both phases are present. It also seems reasonable that in a soil mass having a given temperature and moisture gradient that the potential due to capillary action is opposed by the potential due to vapor pressure. With these two opposed potentials (one must overcome the other) the net moisture diffusion will take place. Whether radioactive techniques are able to detect the vapor phase or not I cannot say. Some study must be made to determine the net potential resulting in a net moisture diffusion.

AUTHORS' CLOSURE (Raymond Eisenstadt): The authors appreciate the comments evoked by this paper and want to note that they are cognizant of the magnitude of the problem. It is recognized that the presentation of this paper is somewhat oversimplified for reasons of clarity.

We admit that the data presented and the conclusions drawn represent an ideal situation and cannot be used for design purposes without further study. Such studies should be made on actual soils. This paper may serve as a starting point. In response to Dr. Hutcheon and Dr. Touloukian let me state that a more adequate list of references is contained in my doctorate thesis.

It should be emphasized that these are experimental results, and are sometimes not as accurate as might be desired. Other typical runs of data were compiled, but not listed in the paper; these perhaps would offer better substantiation of the conclusions, and illustrate how surface adsorption of the cation ion was eliminated.

1485

No. 1485

DESIGN DATA FOR SLAT-TYPE SUN SHADES FOR USE IN LOAD ESTIMATING

By G. V. PARMELEE* AND D. J. VILD**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

A STUDY OF the effectiveness of slat-type sun shades has been carried on at the A.S.H.V.E. Research Laboratory, Cleveland, as part of the Society's research program. It has had the advisory guidance of the Technical Advisory Committee on Heat Flow Through Glass.† This paper gives design data for Venetian blinds and sun screens in combination with several types of single flat glass based upon experimental data[1,2] and a previously presented theoretical treatment.[3] The data in this paper are limited to sun shades which are in a vertical plane, and which have opaque slats running in a horizontal direction set at such angles as to tend to exclude radiation from above-the-horizon sources.

CHARACTERISTICS OF SLAT-TYPE SUNSHADES

The effectiveness of a sun shade can be examined from two viewpoints: (1) the part of the sun's radiant energy that the shade excludes, and (2) the part if the sun's energy that it admits to the room. Because in most applications engineers need to estimate the magnitude of sun heat gains, the problem will be discussed with the second viewpoint in mind.

Heat gain from a sun shade-window combination consists of two components:

1. Heat gain due to transmitted direct and diffuse solar radiation, and
2. Heat gain due to convection and radiation.

The first component refers to the solar energy which passes through the the sun shade; the second component refers to the gain which results from solar

* Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

** Research Engineer, A.S.H.V.E. Research Laboratory.

† *Personnel:* R. A. Miller, *Chairman*, N. B. Hutcheon, *Vice Chairman*, W. J. Arner, F. L. Bishop, E. W. Conover, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, D. R. Muir, H. B. Vincent.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

radiation being absorbed by the glass-shade combination. These heat gains are dependent upon the variables in the following list (also see Figs. 1 and 2):

1. *Position of the sun:* (a) Profile or shadow-line angle, ϕ; (b) Altitude of the sun, β; (c) Incident angle, θ.

2. *Sun shade characteristics:* (a) Color (or solar absorptance, α) of the slats; (b) Slat angle, ψ; (c) Ratio of slat width to slat spacing, W/S, hereafter referred to as spacing ratio; (d) Ratio of slat thickness to slat spacing, D/S, hereafter referred to as thickness ratio.

3. Shade location, *i.e.*, indoors or outdoors.

4. Characteristics of the glass with which the shade is used.

5. Weather conditions and the indoor environment.

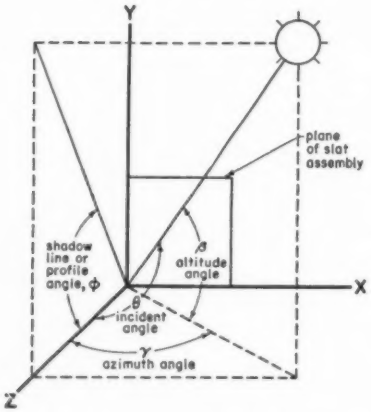


FIG. 1. DEFINITION OF ANGLES

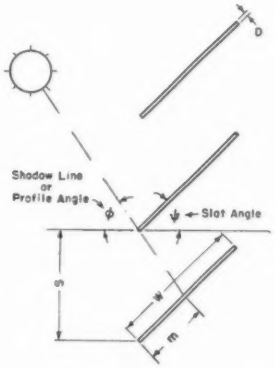


FIG. 2. GEOMETRY OF SLAT-TYPE SUNSHADES

The adjustment or design of any slat-type sun shade for *best performance* requires a compromise between sun heat exclusion, lighting, and vision. If a space is not to be air conditioned, ventilation must also be considered. The possibilities of obtaining adequate natural lighting on sun exposures with a minimum of energy input to the room through the use of slat-type sun shades is an interesting problem that merits investigation. This paper and others[1,2,3] provide some useful data and analytical methods applicable to such problems.

The surfaces of most painted or chemically treated slats of shades in present use can be classed as diffuse reflectors. A general discussion of the performance characteristics of sun shades with this type of slat follows:

1. The spacing ratio of Venetian blinds being manufactured today is standardized at 1.2. The slat angle is adjustable through all angles. Normal slat curvature does not significantly change the shade performance. The thickness ratio for metal slats is so small as to be insignificant.

2. The spacing ratio of screen-type shades of current manufacture varies from 0.8 to 1.0. Slat angles are fixed and most shades have a 17 deg slat angle. Some are set at 30 deg. The thickness ratio is an important factor.

TABLE 1—SHADE FACTORS FOR SLAT-TYPE SUN SHADES WITH DIFFUSE REFLECTING SLATS

SOLAR ABSORPTANCE OF SLAT α	COLOR OF SLAT (approx)	SLAT WIDTH SPACING RATIO, W/S	SLAT ANGLE ψ	SHADE LOCATION	GLASS TYPE	SHADE FACTORS		
						Window in Shade	Window in Sun	
							Factor	Lower Limit of Profile Angle Degrees
0.2	See footnote[a]	1.2[b]	45	Indoors	Common	0.51	0.45	10
0.4	white, ivory cream	1.2[b]	45	Indoors	Common	0.59	0.56	10
0.6	light colors	1.2[b]	45	Indoors	Common	0.67	0.65	10
0.8	dark colors	1.2[b]	45	Indoors	Common	0.75	0.75	10
1.0	black	1.2[b]	45	Indoors	Common	0.81	0.82	10
0.4	white, ivory cream	1.2	0[c]	Indoors	Common	0.77	0.69	40
0.4	"	1.2	30	Indoors	Common	0.65	0.59	20
0.4	"	1.2	45	Indoors	Common	0.59	0.56	10
0.4	"	1.2	60	Indoors	Common	0.53	0.49	none
0.4	"	1.2	90[c]	Indoors	Common	0.37	0.41	none
0.4	"	1.2	45	Indoors	Regular Plate[d]	0.63	0.58	10
0.4	"	1.2	45	Indoors	Heat Absorbing Plate[e]	0.56	0.55	10
0.4	white, ivory cream	1.2	45	Outdoors	Common	0.30	0.15	10
0.4	green tint[f]	1.0	17	Outdoors	Common	0.37	0.22	30
0.8	black[g]	0.8	17	Outdoors	Common	0.32	0.15	40

a Approximated by diffuse reflecting aluminum with protective anodic coating.
b Standard spacing ratio for venetian blinds; thickness-spacing ratio negligible.
c Estimated, see text.
d Normal incidence transmittance equals 0.77.
e Normal incidence transmittance equals 0.41.
f Screen-type shade of chemically treated aluminum; thickness ratio = 0.15.
g Screen-type shade of chemically treated bronze; thickness ratio = 0.085.

3. Transmitted solar radiation consists of *straight-through* and *reflected-through* components. Both components are influenced by profile or shadow-line angle, slat angle, and spacing ratio. The *reflected-through* component is also dependent upon the absorptance of the slat for solar radiation and does not change rapidly with profile angle.

4. An increase in slat angle increases the total energy reflected to the outside and decreases the amount admitted to the room.

5. A spacing ratio of 1.2 with a slat angle of 45 deg will exclude the *straight-through* component on all orientations in the north latitudes between 6 a.m. and 6 p.m. from May 1 to the middle of August. A 30-deg slat angle will do the same between 7 a.m. and 5 p.m. but will approximately double the *reflected-through* component and increase the radiation absorbed by the shade.

6. The slat surface absorptance of most of the Venetian blinds manufactured today is of the order of 0.40 for total solar energy. The absorptance for light is considerably less.

7. A decrease in slat absorptance (higher slat reflectance) increases the amount of solar radiation admitted to the room by increasing the *reflected-through* component,

TABLE 2[a]—INSTANTANEOUS RATES OF HEAT GAIN DUE TO TRANSMITTED DIRECT AND DIFFUSE OR SKY SOLAR RADIATION BY A SINGLE SHEET OF UNSHADED COMMON WINDOW GLASS

For Clear Atmospheres and 18 Deg Declination, North (August 1)

Note: For total instantaneous heat gain, add these values to the Table 3 Values.

LATITUDE	SUN TIME A.M. → ↓		INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)								
			N	NE	E	SE	S	SW	W	NW	Horiz
	6 a.m.	6 p.m.	25	98	108	52	5	5	5	5	17
	7	5	23	155	190	110	10	10	10	10	71
	8	4	16	148	205	136	14	13	13	13	137
30 Deg N.	9	3	16	106	180	136	21	15	15	15	195
	10	2	17	54	128	116	34	17	16	16	241
	11	1	18	20	59	78	45	19	18	18	267
	12		18	19	19	35	49	35	19	19	276
	5 a.m.	7 p.m.	3	7	6	2	0	0	0	0	1
	6	6	26	116	131	67	7	6	6	6	25
	7	5	16	149	195	124	11	10	10	10	77
40 Deg N.	8	4	14	129	205	156	18	12	12	12	137
	9	3	15	79	180	162	42	14	14	14	188
	10	2	16	31	127	148	69	16	16	16	229
	11	1	17	18	58	113	90	23	17	17	252
	12		17	17	19	64	98	64	19	17	259
	5 a.m.	7 p.m.	20	54	54	20	3	3	3	3	6
	6	6	25	128	149	81	8	7	7	7	34
	7	5	12	139	197	136	12	10	10	10	80
50 Deg N.	8	4	13	107	202	171	32	12	12	12	129
	9	3	14	54	176	183	72	14	14	14	173
	10	2	15	18	124	174	110	16	15	15	206
	11	1	16	16	57	143	136	42	16	16	227
	12		16	16	18	96	144	96	18	16	234
		↑ P.M. →	N	NW	W	SW	S	SE	E	NE	Horiz

[a] Same as Table 13, Chapter 13, THE GUIDE 1953.

TABLE 3[a]—INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM A SINGLE SHEET OF UNSHADED COMMON WINDOW GLASS

For Clear Atmospheres and 18 Deg Declination, North (August 1)
For 80 F Indoor Temperature

Note: For total instantaneous heat gain, add these values to the Table 2 values.

SUN TIME	DRY-BULB DEG F	NORTH LATITUDE DEGREES	INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)								
			N	NE	E	SE	S	SW	W	NW	Horiz
5 a.m.	74		−6	−6	−6	−6	−6	−6	−6	−6	−6
6	74		−5	−4	−4	−5	−5	−6	−6	−6	−5
7	75		−5	−2	−2	−3	−5	−5	−5	−5	−3
8	77		−3	0	1	0	−2	−3	−3	−3	0
9	80		0	2	4	3	1	0	0	0	3
10	83		3	4	6	6	5	3	3	3	8
11	87		8	8	10	11	10	9	8	8	13
12	90	30, 40, 50	12	12	12	13	14	13	12	12	16
1 p.m.	93		15	15	15	16	17	17	17	15	20
2	94		16	16	16	16	18	19	19	17	21
3	95		17	17	17	17	19	21	21	19	21
4	94		16	16	16	16	17	20	20	19	19
5	93		15	15	15	15	15	18	19	18	17
6	91		13	13	13	13	13	14	15	15	13
7	87		8	8	8	8	8	8	8	8	8
8	85		6	6	6	6	6	6	6	6	6
9	83		3	3	3	3	3	3	3	3	3

[a] Same as Table 14, Chapter 13, THE GUIDE 1953.

but it also increases the energy reflected to the outside. The total heat gain is therefore reduced.

8. Roughly ½ of the *reflected-through* component enters the room in a generally upward direction, if the source of radiation is from above the horizon.

9. Slat-type shades have a high transmittance for ground reflected solar radiation, which may constitute a sizeable fraction of the incident diffuse solar radiation. The transmittance for both *above-the-horizon* and *below-the-horizon* diffuse solar radiation is generally greater than the reflected-through transmittance of direct solar radiation.

10. When a shade is installed on the outdoor side of common glass the convection and radiation gain is equal to about 5 percent of the solar radiation absorbed by the shade-glass combination plus the convection and radiation gain for single unshaded common glass under the same conditions of temperature and weather. Slat size and slat angle variations between 20 to 45 deg have only a small effect.

11. When a shade is installed on the indoor side of common glass, the percentage is about 75.

12. A shade installed indoors reduces the overall steady state conductance of a single window. For 30 deg slats with a spacing ratio of 1.2, the reduction is about 15 percent.

Although discussion has been confined to shades which have diffuse reflecting slats, those interested in the problem should not overlook the high reflectance[1] of specular reflecting (mirror like) surfaces under certain conditions.

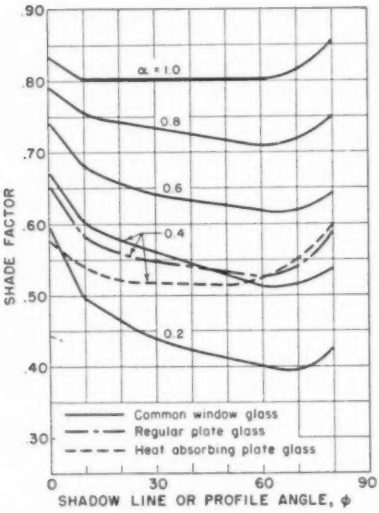


FIG. 3. SHADE FACTORS FOR INDOOR SLAT-TYPE SUNSHADES *vs.* PROFILE ANGLE, ϕ, AND SLAT ABSORPTANCE, α: SLAT ANGLE, $\psi = 45$ DEG; SPACING RATIO, $W/S = 1.2$

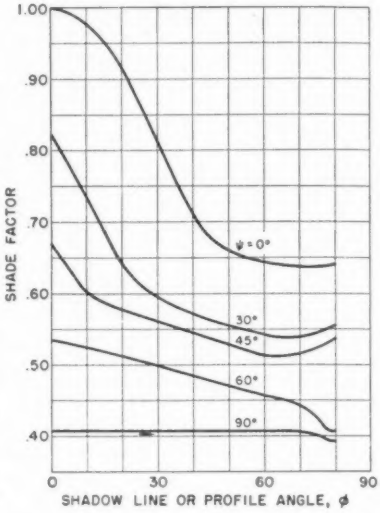


FIG. 4. SHADE FACTORS FOR INDOOR SLAT-TYPE SUNSHADES *vs.* PROFILE ANGLE, ϕ, AND SLAT ANGLE, ψ: SLAT ABSORPTANCE, $\alpha = 0.4$; SPACING RATIO, $W/S = 1.2$

In concluding this discussion, it is well to emphasize that the principles and data given in this and earlier papers have general application not only to the common Venetian blind but also any sun screen of flat opaque slats.

DESIGN DATA

Shade Factors: The shade factor is defined as the total gain from a shade-glass combination minus the convection and radiation gain from single *unshaded common window glass* divided by the total incident solar radiation transmitted by single unshaded common window glass. In equation form,

$$\text{Shade Factor} = \frac{\text{Total Gain from Shade-Glass Combination} - \text{Convection \& Radiation Gain from Single Unshaded Common Glass}}{\text{Total Solar Energy Transmitted by Single Unshaded Common Glass}}$$

Design shade factors for a number of combinations are listed in Table 1. The total gain for a shaded window is found by multiplying the values of Table 2 by the appropriate shade factor and adding to these values the corresponding values from Table 3, (Table 2 gives total transmitted solar radiation and Table 3 gives convection and radiation gain from single unshaded common window

glass.) Procedures and data by which the performance of other shade-glass combinations can be estimated are given in Appendixes A, B and C.

The values for sunlit windows are average values based upon calculations for several orientations. Actually the shade factor varies somewhat with orientation and time of day, because the proportion of direct to diffuse incident solar radiation as well as the profile angle varies. However, for the range in angles indicated for each shade, this variation is small enough to be ignored in estimating cooling load. The lower limit of the profile angle for which each value is applicable is the approximate *cut-off* angle. For angles less than this value

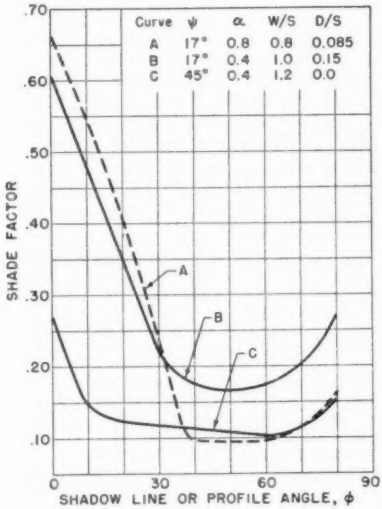


FIG. 5. SHADE FACTORS FOR OUTDOOR SLAT-TYPE SUNSHADES *vs.* PROFILE ANGLE, ϕ

the sun's direct rays can pass through the slats and the shade factor increases rapidly. This is offset in part by the fact that below about 15 deg a decrease in profile angle is accompanied by a decrease in sun intensity. This results in a moderate increase in heat gain, or a decrease, depending upon orientation.

The factors of Table 1 which have a lower limit of 10 deg cover most cases. To supplement Table 1, Figs. 3, 4, and 5 give shade factors as a function of the shadow line or profile angle. The factors given by these figures apply to direct solar radiation only so that computations must treat direct and diffuse radiation separately. These figures are to be used with Table 4, which lists profile angles for three latitudes and eight orientations on August 1. The use of these factors is illustrated in a following section.

Shade Color: Shade factors have been given in terms of total solar energy absorptance of the slat surface. This value can be established by tests in sun-

light or by calculation from spectral absorptance or reflectance values. While spectral data are readily obtainable for the visible portion of the sun's spectrum, data for the infra-red portion of the spectrum are rarely available and hence the total absorptance usually must be estimated. From a practical application viewpoint, factors must be selected on the basis of color. Hence, colors

TABLE 4—VALUES OF THE SHADOW LINE OF PROFILE ANGLE, ϕ, FOR VARIOUSLY ORIENTED VERTICAL SURFACES

Computed for 18 Deg Declination, North (August 1)

LATITUDE	SUN TIME A.M. → ↓		PROFILE ANGLE, ϕ, DEGREES N	NE	E	SE	S	SW
	6 a.m.	6 p.m.	30	10	9	18		
	7	5	68	26	22	34		
	8	4	87	43	35	45		
30 Deg N.	9	3		60	48	54	85	
	10	2		75	61	63	80	
	11	1		88	76	72	79	
	12				90	81	78	81
	5 a.m.	7 p.m.	1	1	1	2		
	6	6	40	13	12	22		
	7	5	75	29	23	34		
40 Deg N.	8	4		47	35	42	83	
	9	3		65	47	49	75	
	10	2		81	60	57	71	
	11	1			75	65	69	86
	12				90	74	67	74
	5 a.m.	7 p.m.	11	4	4	12		
	6	6	49	16	14	24		
	7	5	90	32	24	32	90	
50 Deg N.	8	4		50	34	38	72	
	9	3		69	45	44	65	
	10	2		87	58	50	61	
	11	1			74	57	59	78
	12				90	66	58	66
	↑ P.M. →		N	NW	W	SW	S	SE

have been included in Table 1, but this classification is approximate. About 80 percent of the Venetian blinds manufactured today have white, ivory or eggshell finish. Light absorptances[4] of a typical sample of each were 18, 30 and 34 percent respectively. However, for the total solar spectrum the absorptance values would be higher. Tests[1,5] of white painted surfaces in sunlight gave values ranging from 0.28 to 0.51. Tests with artificial light sources[6,7] gave values of from 0.18 to 0.32 for commercial white paints with values[7] for white pigments in oil as high as 0.43. The temperature rise[8] of glossy white, flat white, ivory and canary yellow paints exposed to sunlight were 31, 35, 40, and 45 percent, respectively, of the rise observed for lampblack under the same conditions. Approximate solar absorptances would be a few percentage points higher in each case. It is evident, therefore, that the selection of shade factors in terms of slat color is an approximation. It is also apparent that careful

selection of paints to be used on indoor Venetian blinds can lead to significant improvements in shade performance. From the viewpoint of sun heat reduction, color does not seem to be important for outdoor shades. It may assume importance if the shade has light control functions.

ILLUSTRATIVE EXAMPLES

Example 1: Find the total instantaneous rate of heat gain for common window glass provided with an indoor white Venetian blind. Slats are set at 45 deg and have a spacing ratio of 1.2. The orientation is west, the time is 4 p.m. on August 1 and the latitude is 40 deg, north. Design dry bulb temperatures are 80 F indoor and 95 F maximum outdoor.

Solution: Table 4 shows that the profile angle is 35 deg. Hence, factor 0.56 of Table 1 is applicable. Tables 2 and 3 show that the gain due to transmitted solar radiation and to convection and radiation are, respectively, 205 and 20 Btu per (hr) (sq ft) for unshaded common window glass, giving a total gain of 225 Btu per (hr) (sq ft). The total gain for the shaded window equals

$$0.56\ (205) + 20 = 134.8 \text{ Btu per (hr) (sq ft)}$$

Example 2: Same as *Example 1* except the slats are horizontal (slat angle = 0 deg).

Solution: The Table 1 factor of 0.69 cannot be used because the profile angle is less than 40 deg (Table 4). The gain must be found in three steps. The gain due to diffuse radiation on an unshaded west window at 4 p.m. is 28 Btu per (hr) (sq ft). [See Chaper 13, THE GUIDE 1953. For a solar altitude at 4 p.m. of 34.5 deg (GUIDE Table 6), the incident diffuse radiation is 36 Btu per (hr) (sq ft) (GUIDE Table 4) of which 0.79 (GUIDE Table 12) is transmitted.] The gain due to direct radiation on the west window is therefore $205 - 28 = 177$ Btu per (hr) (sq ft). The shade factors are 0.77 for diffuse radiation (Table 1) and 0.76 for direct radiation (Fig. 4). The total instantaneous gain for the unshaded window equals

$$0.77\ (28) + 0.76\ (177) + 20 = 176.1 \text{ Btu per (hr) (sq ft)}$$

Example 3: Same as *Example 1* except that the glass is single heat absorbing plate glass.

Solution: Note that the factor of 0.55 in Table 1 is applied to the transmitted gain of *single common window glass* and that Table 2 values are used as in the preceding examples. (The factors of Table 15 and the values of Table 16 in Chapter 13 of THE GUIDE 1953 lead to a total gain of 150.3 Btu per (hr) (sq ft) for single unshaded heat absorbing glass, but are not used in the computations for shading.) The total instantaneous gain for shaded heat absorbing glass is

$$0.55\ (205) + 20 = 132.8$$

Example 4: Same as *Example 1* except that the window has a screen-type shade with black slats on outside of window. The spacing ratio is 0.8, the slat angle is 17 deg.

Solution: This example is worked like *Example 2.* The shade factors are 0.32 (Table 1) and 0.15 (Fig. 5) for diffuse and direct radiation respectively. The total instantaneous gain is

$$0.32\ (28) + 0.15\ (177) + 20 = 55.5 \text{ Btu per (hr) (sq ft)}$$

DESIGN ASSUMPTIONS

Experimental work showed that the increase in convection and radiation gain caused by the addition of an indoor shade to common glass was proportional to the solar radiation absorbed by the combination and was practically independent of slat width (0.049 and 2 in.), spacing ratio (0.8 and 1.2), and slat angle (20 to 45 deg) values covered by the experiments. It seemed reasonable to conclude that the same proportional value could be applied to both the horizontal and closed slat positions without serious error.

Night tests run with an indoor white shade with slats at 30 deg and spacing ratio of 1.2 showed that this arrangement reduced the overall coefficient of heat transfer U by 15 percent. However, with sun incident on the window, the effect of differences between the indoor and outdoor temperatures was not apparent in the experiments. Hence, no adjustment has been made to the values in Table 3 (convection and radiation gain from common unshaded glass). The shade factors of Table 1 and Figs. 3, 4, and 5 can therefore be used for all seasons and orientations with reasonable accuracy. It is only necessary to compute the gains due to transmitted solar radiation by unshaded common window glass and the gain (or loss) by convection and radiation and to proceed as in *Example 2.*

The effect of supporting tapes has been ignored. They would reduce the transmitted gain by 5 to 8 percent. The shade absorptance would be altered, depending upon color relative to the slat color.

Diffuse radiation comes from the sky and also from the ground by reflection of both direct and diffuse radiation. Analysis of test data showed that about 25 percent of the incident diffuse radiation came from the ground and 75 percent from the sky. These proportions were used in computing design values of absorptance and transmittance of diffuse solar radiation.

Allowance should be made if necessary for partial shading by window reveals and sash members.

DISCUSSION

Comparison of the shade factors of Table 1 with values given in Table 24 of Chapter 13 of THE GUIDE 1953 can be made if light, medium and dark colors be accepted as equivalent to slat absorptances of 0.4, 0.6 and 0.8 respectively. For a 45-deg slat setting and a spacing ratio of 1.2 the Table 1 values are 0.56, 0.65 and 0.75 compared with the values of 0.62, 0.74, and 0.86 for the three colors respectively in THE GUIDE. The new data give about a 10 percent greater reduction in sun load than the presently used data. The new factor for white slats outside, 0.15, is just one-half the value of 0.30 in THE GUIDE. The new values for the black screen-type shade are also lower. The value of 0.70 given for aluminum slats, in THE GUIDE, also needs revision. The specific reflecting character of the aluminum is not known, but both specular and diffuse reflecting aluminum should give much lower values, although the factor for specular slats could be this high for low profile angles.

Table 1 and Fig. 4 show that the effectiveness of a slat-type shade increases as the slat angle increases. A minimum factor of 0.41 is attained by the shade with slat absorptance of 0.4 when the slats are fully closed. This value is, as indicated previously, an estimate. The values for a closed blind, a fully drawn

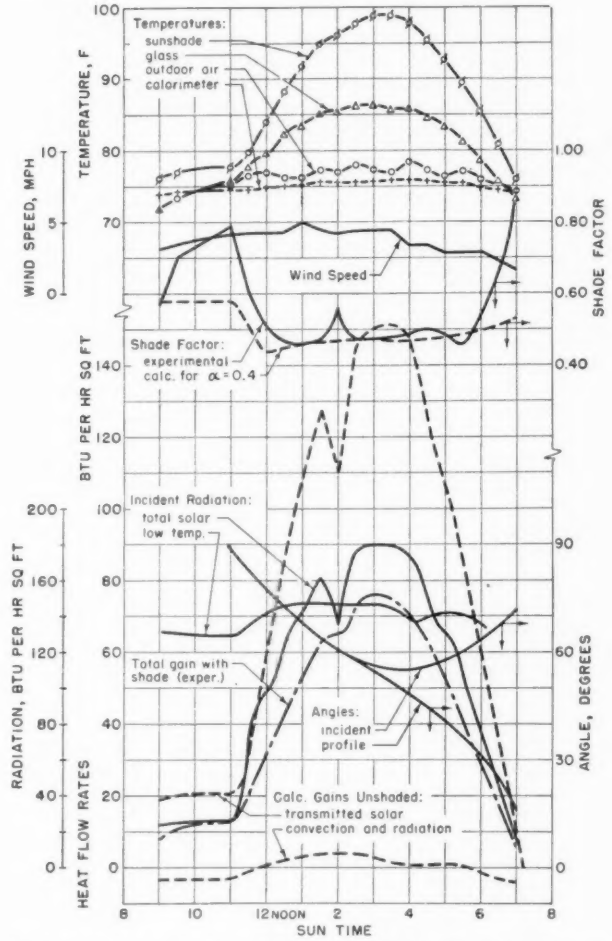


FIG. 6. TEST RESULTS AND SHADE FACTORS FOR WHITE SLAT SHADE INSIDE GLASS NO. 2: SOUTHWEST ORIENTATION, JULY 2, 1951, SLAT ANGLE—45 DEG, SLAT SPACING RATIO—1.2

roller shade, and a drawn drape, should be about the same if each has the same absorptance for solar radiation. The value of 0.45 used in THE GUIDE for a fully drawn white roller shade (it is believed that this is an estimate based upon calculations) compares favorably.

Table 1 shows that outdoor shades are very effective and that it makes little difference whether the exclusion of sun heat is by reflection or by absorption

with heat loss to the outdoor environment by convection and reradiation. Note, however, from Fig. 5 that the screen-type shade with fixed slats decreases rapidly in effectiveness as the profile angle decreases, starting at about 35 to 40 deg. Below about 15 deg, this is compensated in part by the fact that sun intensity decreases, so that heat gains increase less rapidly than the effectiveness decreases, or they may decrease, depending upon orientation.

It will be seen that total heat gain with a heat absorbing glass with an indoor white shade is practically the same as for common window glass and a white shade. Compared with unshaded heat absorbing glass, the total gain is reduced about 10 percent by the use of a shade. These findings are similar to those observed by Leopold[9] in scale model tests with an artificial sun. Table 1 gives factors of 0.56, 0.58, and 0.55 for different types of flat glass with an indoor shade having a slat absorptance of 0.4, a slat angle of 45 deg and a spacing ratio of 1.2. The small difference in these values indicates that the factor for any given shade and common window glass is a reasonably satisfactory value for this shade with any type of single flat glass.

Data on glass and shade temperatures to be expected under various conditions and a detailed analysis of the heat gain components are given in Reference 2. Typical values of temperature are illustrated by Fig. 6 which gives test data for a white shade on the inside of a high transmission glass (sample No. 2, see Reference 10). The slat angle was 45 deg and the spacing ratio was 1.2. The observed solar intensity values are slightly below design values and the outdoor dry-bulb temperatures are much below, a combination of weather conditions that is fairly typical of the Cleveland area. The low wind velocity is typical of the conditions under which most of the experimental work was done.

Dotted curves give the calculated gain due to transmitted solar radiation and the calculated gain due to convection and radiation from unshaded glass No. 2. The third dotted curve gives theoretical shade factors for a slat absorptance of 0.4. These values are compared with values based upon the experimental observations. Test values are close to the calculated values during the most important part of the day, but high at the beginning and end, when a few Btu per (hr) (sq ft) make a large percentage difference. Differences can be attributed to experimental error, to the fact that heat storage rates of glass and shade were not taken into account in computing the theoretical shade factors, and other factors.

It must be emphasized that the *heat gain values are instantaneous rates of heat gain.* In practical situations, *heat storage of the internal structure must be considered* in determining the contribution of solar heat gains to cooling system load.

REFERENCES

1. A.S.H.V.E. RESEARCH REPORT No. 1474—The Shading of Sunlit Glass—An Experimental Study of Slat-Type Sun Shades, by G. V. Parmelee, W. W. Aubele and D. J. Vild (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 221).

2. A.S.H.V.E. RESEARCH REPORT—The Shading of Sunlit Glass—Convection and Radiation Gain from Slat-Type Sun Shades, by G. V. Parmelee and D. J. Vild (to be published).

3. A.S.H.V.E. RESEARCH REPORT No. 1460—The Shading of Sunlit Glass—An Analysis of the Effect of Uniformly Spaced Flat Opaque Slats, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 377).

4. Private Communication from J. A. Meacham (Sherwin-Williams Co., Cleveland, Ohio, March 1953).

5. Effective Solar Absorption of Various Colored Paints, by R. H. Heilman and R. W. Ortmiller (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 399).

6. Reflection Coefficients of Materials for Solar Radiation (British Building Research Board, Report for 1932, *H.M.S. Stationery Office,* London, 1933, p. 87).

7. A New Means of Minimizing Radiant Heat from High Level Lighting Systems, by D. Cannell, E. Q. Adams, and S. C. Forbes (*I.E.S. Transactions,* Vol. 34, No. 7, July 1939, p. 726).

8. Solar Heating of Various Surfaces, by H. V. Cottony and R. S. Dill (*National Bureau of Standards Report BMS64,* Washington, D. C., January 23, 1941).

9. The Mechanism of Heat Transfer, Panel Cooling and Heat Storage—Part II—Solar Radiation, by C. S. Leopold (*Refrigerating Engineering,* June 1948, p. 571).

10. A.S.H.V.E. RESEARCH REPORT No. 1333—Measurements of Solar Heat Transmission Through Flat Glass, by G. V. Parmelee, W. W. Aubele, and R. G. Huebscher (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 165).

APPENDIX A

Procedure in the Calculation of Transmittance, Absorptance and Reflection of Radiation by Shades with Diffuse Reflecting Slats

Transmittance and absorptance curves for slat width-spacing ratios of 0.6 and 1.2 and slat angles of 0, 30 and 45 deg were given in a recent research paper (Reference A-1). These curves were for positive profile angles only. It frequently is necessary to determine the performance of shades with other slat angles and spacing ratios or for negative profile angles. Moreover, it may be necessary to know approximately how the radiation which passes through a shade is distributed. This information is needed in estimating the radiant energy input to different room surfaces, some of which may be cooled by cooling panels, and also in analyzing the control of daylight by louver devices. The purpose of this Appendix is to present the information needed for such estimates. After the basic equations were published, more complete data on configuration factors were developed and additional transmittance and absorptance values were computed.

The equations are based on the following idealized conditions:

1. Slat surfaces are perfectly reflecting diffuse surfaces.
2. Surface absorptance is independent of incident angle.
3. Slats are flat, opaque, infinitely long and the edges are square.

Practical slats approximate these conditions reasonably well. Normal curvature has little effect and the requirement of infinite length is closely met by a slat length-width ratio of 10 or greater.

TERMINOLOGY

The principal terminology is illustrated by Figs. 1 and 2 of the paper. The shadow line or profile angle, ϕ, is the sun angle which controls the performance of the shade. The relationship between this angle, the solar altitude, β, and the solar azimuth, γ, is as follows:

Length of slat horizontal, plane of assembly vertical (most common arrangement):

$$\tan \phi_{v,h} = \tan \beta / \cos \gamma \quad \text{(A-1)}$$

Length of slat horizontal, plane of assembly inclined at angle, ω, as in a jalousie:

$$\phi_{s,h} = \phi_{v,h} \pm \omega \quad \text{(A-2)}$$

Length of slat vertical, plane of assembly vertical:

$$\phi_{v,v} = \gamma \quad \text{(A-3)}$$

Length of slat inclined and plane of assembly inclined at angle, ω:

$$\tan \phi_s = (\cos \phi_{v,h} \tan \gamma)/\cos \phi_{s,h} \quad \text{(A-4)}$$

PART I—TRANSMITTANCE OF A BEAM OF RADIATION

The component parts of the equations for transmittance as originally published are given herein in order that the approximate directional distribution can be determined. Changes in the designation of configuration factor, F, have been made in order to distinguish between those which must take into account the non-uniform distribution of energy reflected from a slat surface and those which do not. The method of taking slat thickness into account is treated in a later paragraph. The equations are applicable to slats of which both surfaces have the same radiant energy absorptance. The case where opposite surfaces have different absorptances is treated in Part VI.

The equations are strictly applicable to monochromatic radiation incident upon a shade whether it is short wavelength radiation from the sun or long wavelength radiation from sources at room temperature. Suggestions for treating cases where the slat surface absorptance, α, varies with wavelength are given in Part IV. Examples in the use of the equations are given in the last section of this appendix.

The fraction of an incident beam of radiation which is transmitted by the shade alone is

$$T_D = L + X + Y + Z \quad \text{(A-5)}$$

where

L, X, Y and Z are components defined and found as follows:

Straight-through Component, L: This fraction passes through without touching any slat surface

$$L = 1 \pm \frac{W}{S} \frac{\sin (\phi + \psi)}{\cos \phi} \quad \text{(A-6)}$$

Reflected-through Component, X: This fraction of the beam passes through the slat systems after one reflection from the slat surface which is directly irradiated by the sun.

$$X = (1 - L)\left[F_1 (1 - \alpha)\right] \quad \text{(A-7)}$$

Reflected-through Component, Y: This fraction leaves the slat surface opposite to the surface directly irradiated by the sun after undergoing an infinite series of reflections.

$$Y = (1 - L)\left[\frac{F_2 F_3 (1 - \alpha)^2}{1 - F_4^2 (1 - \alpha)^2}\right] \quad \text{(A-8)}$$

Reflected-through Component, Z: This fraction leaves the slat surface which is directly irradiated by the sun after undergoing an infinite series of reflections.

$$Z = (1 - L)\left[\frac{F_2 (1 - \alpha)^2}{1 - F_4^2 (1 - \alpha)^2}\right]\left[F_3' F_4 (1 - \alpha)\right] \quad \text{(A-9)}$$

TABLE A-1—CONFIGURATION FACTORS F_1[a]

SLAT ANGLE, ψ DEG	W/S	m/W = FRACTION OF SLAT WIDTH IRRADIATED				
		0	¼	½	¾	1
0	0.6	0.243	0.269	0.296	0.328	0.362
	0.8	0.186	0.212	0.244	0.282	0.324
	1.0	0.146	0.169	0.202	0.243	0.292
	1.2	0.116	0.138	0.168	0.211	0.264
	1.4	0.094	0.114	0.142	0.184	0.242
15	0.6	0.166	0.183	0.204	0.228	0.255
	0.8	0.130	0.148	0.169	0.196	0.228
	1.0	0.104	0.119	0.140	0.168	0.205
	1.2	0.083	0.098	0.118	0.146	0.186
	1.4	0.063	0.081	0.100	0.128	0.170
30	0.6	0.108	0.119	0.134	0.149	0.167
	0.8	0.082	0.094	0.109	0.127	0.149
	1.0	0.066	0.075	0.090	0.109	0.133
	1.2	0.055	0.063	0.076	0.095	0.121
	1.4	0.048	0.054	0.064	0.083	0.111
45	0.6	0.059	0.066	0.074	0.084	0.095
	0.8	0.046	0.052	0.060	0.071	0.085
	1.0	0.037	0.042	0.050	0.061	0.075
	1.2	0.031	0.035	0.042	0.053	0.068
	1.4	0.027	0.030	0.036	0.046	0.062
60	0.6	0.026	0.029	0.033	0.038	0.043
	0.8	0.020	0.023	0.027	0.032	0.038
	1.0	0.016	0.019	0.022	0.027	0.034
	1.2	0.014	0.016	0.019	0.023	0.031
	1.4	0.012	0.014	0.016	0.020	0.028

[a] For $+\phi$ values and for $-\phi$ values such that $-(-\phi)$ is less than ψ (See Fig. A-1)

If a knowledge of the distribution is not needed, Equations A-8 and A-9 can be combined as follows:

$$(1 - L)\left[F_2\, C_1\, (1 - \alpha)\right] \quad \ldots\ldots\ldots\ldots \quad \text{(A-10)}$$

The Configuration Factors: Configuration factors F are given in Tables A-1 through A-5. These are functions of the fraction, m/W, of the slat width irradiated by the radiant beam which falls on the slat system, of the slat width-spacing ratio, W/S, and of the slat angle, ψ. Since these quantities are dimensionless, they apply to systems of slats of any size and hence may have utility beyond the scope of this paper. Evaluation of the term, m/W, is carried out as described in a later paragraph. As noted under *Rules*, the configuration factor to be used depends upon the sign of the profile angle, ϕ, and its value relative to the slat angle, ψ. For convenience in calculations, values of C_1 (and C_1' for negative profile angles) are given in Tables A-6 and A-7. In carrying out calculations, it may be found desirable to plot the configuration factors against slat angle to facilitate interpolation.

The significance of the configuration factors is explained in part by Fig. A-1. In addition, it should be noted:

1. Radiant energy in a beam from an external source, such as the sun, uniformly irradiates the surface on which it falls.

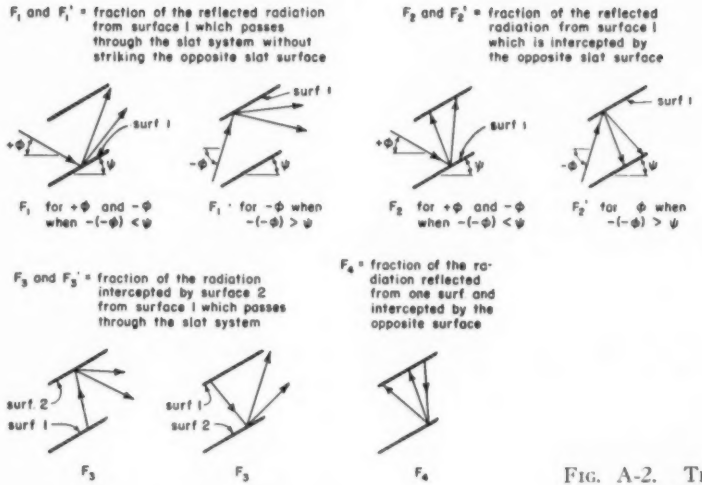

F1 and F1' = fraction of the reflected radiation from surface 1 which passes through the slat system without striking the opposite slat surface
F2 and F2' = fraction of the reflected radiation from surface 1 which is intercepted by the opposite slat surface
surf 1
+Φ
ψ
F1 for +Φ and −Φ when −(−Φ) < ψ
−Φ
F1' for −Φ when −(−Φ) > ψ
F2 for +Φ and −Φ when −(−Φ) < ψ
F2' for Φ when −(−Φ) > ψ
F3 and F3' = fraction of the radiation intercepted by surface 2 from surface 1 which passes through the slat system
F4 = fraction of the radiation reflected from one surf and intercepted by the opposite surface
surf 2
surf 1
F3
F4

FIG. A-1. EXPLANATION OF CONFIGURATION FACTORS

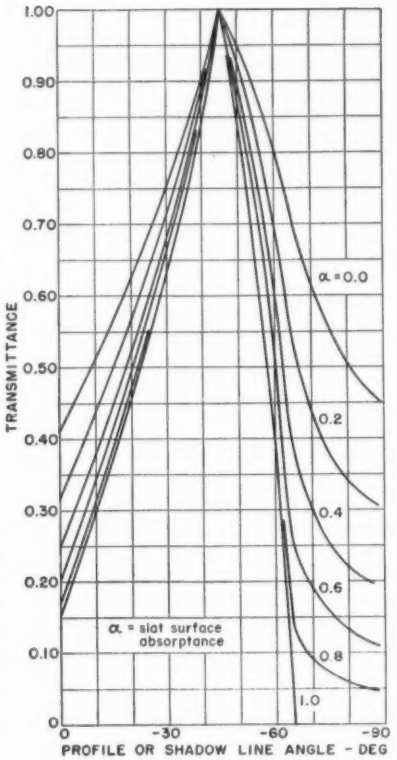


FIG. A-2. TRANSMITTANCE OF DIFFUSE REFLECTING SLATS FOR MONOCHROMATIC RADIATION *vs.* SLAT SURFACE ABSORPTANCE FOR NEGATIVE PROFILE ANGLES AND SLAT ANGLE $\psi = 45$ DEG; $W/S = 1.2$

2. Radiant energy reflected from one slat to a second slat is not uniformly distributed over the second slat.

3. For $m/W = 1.0$, $F_2 = F_2'$, but does not equal F_4, since F_4 takes into account approximately the non-uniform distribution of reflected energy which falls on slat surfaces.

4. F_3 and F_3' also takes into account approximately non-uniform distribution of reflected radiant energy.

5. For $m/W = 1.0$, $F_1 + F_1' + F_2 = 1.0$

6. $F_3 + F_3' + F_4 = 1.0$

These configuration factors were computed from data and methods contained in Reference A-2.

The following Rules should be observed:

1. For $+\phi$ values or when $-(-\phi)$ is less than ψ:
   (a) Use $-$ sign in Equation A-6.
   (b) When L becomes negative, the straight through component is zero. Therefore, in Equations A-7, A-8, A-9 and A-10 L is assigned a value of zero.
   (c) See Tables A-1, A-2, A-5 and A-6 for values of the factors F_1, F_2, F_3, F_3', F_4 and C_1.

2. For ϕ values such that $-(-\phi)$ is greater than ψ:
   (a) Use $+$ sign in Equation A-6.
   (b) When L becomes negative, the straight through component is zero and in Equations A-7, A-8, A-9 and A-10 L is assigned a value of zero.
   (c) Replace F_1, F_2, and C_1 by F_1' F_2', and C_1' and replace F_3 in Equation A-8 with F_3' and replace F_3' in Equation A-9 with F_3.
   (d) See Tables A-3, A-4, A-5 and A-8 for configuration factors F_1', F_2', F_3, F_3', F_4 and C_1'.

   Figs. A-2 and A-3 give transmittance values for negative profile angles for some common shade geometries. These will be found to have value in treating ground reflected solar radiation (see Part III which follows). They also supplement values given for positive profile angles in Reference 3 of the paper and facilitate studying the performance of slat type shades in which the long dimension of the slats is vertical. Recall that Equation A-3 of this appendix states that the profile angle is, in such cases, equal to the solar azimuth. In this case, the sun's position will lead to both positive and negative profile angles.

3. Configuration factors F_1, F_2, etc., depend upon the fraction of the slat width irradiated, m/W (see Fig. 2 of the paper):

$$\frac{m}{W} = 1/(1 - L) \text{ but cannot exceed } 1.0 \quad \ldots\ldots\ldots \quad \text{(A-11)}$$

In computing m/W, L is evaluated according to rules 1(a) and 2(a), and is used in Equation A-11 with the sign and value as given by Equation A-6.

4. Corrections for slat thickness should be considered. In the case of most metal Venetian blinds, the slat thickness-spacing ratio is so small that corrections can be ignored. This is usually not true in the case of screen type shades. The term W/S must be corrected by multiplying by $1/(1\text{-}P)$ where:

$$P = \frac{D/S \cos(\phi + \psi)}{\cos \phi} \quad \ldots\ldots\ldots\ldots \quad \text{(A-12)}$$

Transmittance as determined by using Equations A-6, A-7, A-8 and A-9 with the correction to W/S for thickness ratio applied, is then corrected by multiplying by $(1 - P)$. When $\phi = 90 - \psi$ and ϕ is positive, no correction is required for slat thickness. The configuration factors to be used are those for W/S multiplied by $1/(1 - P)$. These are not exact but no refinement seems justified.

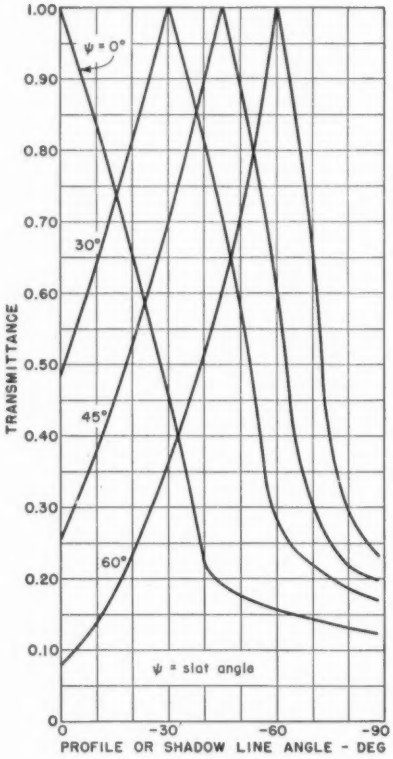


FIG. A-3 *(left)*. TRANSMITTANCE OF DIFFUSE REFLECTING SLATS FOR MONOCHROMATIC RADIATION *vs.* SLAT ANGLE FOR NEGATIVE PROFILE ANGLES AND SLAT SURFACE ABSORPTANCE $\alpha = 0.4$; $W/S = 1.2$

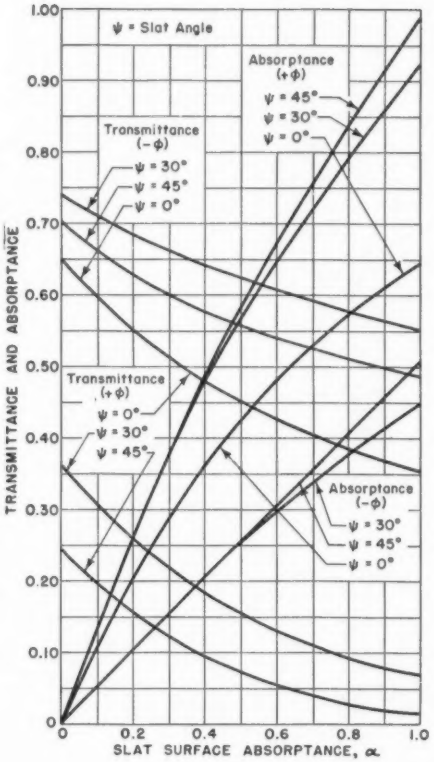


FIG. A-4 *(right)*. TRANSMITTANCE AND ABSORPTANCE OF SLAT-TYPE SUN SHADES FOR RADIATION FROM UNIFORM DIFFUSE SOURCES ABOVE THE HORIZON $(+\phi)$ AND BELOW THE HORIZON $(-\phi)$; $W/S = 1.2$

NOMENCLATURE

α = absorptance of slat surface for solar radiation, dimensionless.
α_D, α_d = absorptance of a glass section for direct and diffuse solar radiation respectively, dimensionless.
β = altitude of the sun above a horizontal plane, degrees.
γ = azimuth of the sun with respect to the perpendicular to a vertical wall, degrees.
λ = wavelength.
θ = angle of incidence; angle between the sun's rays and a perpendicular to a surface, degrees.
$\rho = 1 - \alpha$ = reflectance of a slat surface for solar radiation, dimensionless.
ρ_D, ρ_d = reflectance of a glass section for direct and diffuse solar radiation respectively, dimensionless.
τ_D, τ_d = transmittance of a glass section for direct and diffuse solar radiation, respectively, dimensionless.
ϕ = profile angle, degrees.
ψ = slat angle, degrees.
ω = inclination angle of the plane of the slat assembly measured with respect to a vertical surface, degrees.
A_D, A_d = absorptance of a slat assembly for direct and diffuse solar radiation respectively, dimensionless.
D = slat thickness, inches.
F_1, F_2, F_3, F_4 = configuration factors (see Fig. A-1).
m = slat width irradiated by the sun, inches.
R_D, R_d = reflectance of slat assembly for direct and diffuse solar radiation respectively, dimensionless.
S = slat spacing, inches.
T_D, T_d = transmittance of slat assembly for direct and diffuse solar radiation respectively, dimensionless.
W = slat width, inches.
W/S = ratio, slat width to spacing.
D/S = ratio, slat thickness to spacing.

Subscripts D and d refer to direct and diffuse solar radiation, respectively.

Subscripts h, i, and v refer to the position of the long dimension of the slat and the position of the plane of the slat-shade assembly and are respectively horizontal, inclined and vertical.

PART II—ABSORPTANCE OF A BEAM OF RADIATION

$$A_D = \alpha (1 - L) \left[1 + \frac{F_2 (1 - \alpha)}{1 - F_4 (1 - \alpha)} \right] \quad \ldots\ldots \text{(A-13)}$$

The following Rules should be observed:

1. When L is negative, let $L = 0$.

2. For $-\phi$ values such that $-(-\phi)$ is greater than ψ: Replace F_2 by F_2'. If necessary to take into account slat thickness, correct W/S as described in Part I, Rule 4, find A_D, multiply A_D by $(1 - P)$ and add αP to this quantity.

PART III—CALCULATIONS OF TRANSMITTANCE AND ABSORPTANCE OF DIFFUSE RADIATION

The transmittance and absorptance of diffuse radiation are computed by means of Tables A-7 and A-9. Plot curves of T_D and A_D as a function of profile angle. For each pair of coordinates in Table A-7 select the proper value of T_D (or A_D) from the plotted curve. Multiply this value by the appropriate area factor of Table A-9. Do this for each pair of coordinates. Sum each column, sum the columns, and multiply the reasult by 4.0. This gives the diffuse transmittance for a quarter spherical (or an infinite plane parallel to the plane of the shade) source of uniform intensity. If the radiation source is an above-the-horizon source, such as the sky, and the slats are set as shown in Fig. 2 of the paper, the calculations are carried out only with positive profile angle values. However, some radiation is reflected from the ground (a below-the-horizon source) and for this, the usual slat-type shade has a relatively

TABLE A-2—CONFIGURATION FACTORS F_2[a]

SLAT ANGLE, ϕ DEG	W/S	m/W = FRACTION OF SLAT WIDTH IRRADIATED				
		0	¼	½	¾	1
0	0.6	0.260	0.269	0.277	0.280	0.277
	0.8	0.314	0.337	0.352	0.356	0.352
	1.0	0.352	0.387	0.414	0.423	0.414
	1.2	0.384	0.435	0.469	0.482	0.469
	1.4	0.404	0.472	0.515	0.530	0.515
15	0.6	0.204	0.221	0.240	0.254	0.265
	0.8	0.236	0.272	0.300	0.324	0.339
	1.0	0.266	0.314	0.354	0.386	0.403
	1.2	0.288	0.347	0.402	0.442	0.459
	1.4	0.304	0.379	0.444	0.490	0.506
30	0.6	0.144	0.161	0.182	0.203	0.226
	0.8	0.165	0.197	0.233	0.267	0.300
	1.0	0.184	0.226	0.276	0.325	0.366
	1.2	0.198	0.252	0.317	0.380	0.427
	1.4	0.208	0.273	0.356	0.429	0.475
45	0.6	0.090	0.107	0.120	0.141	0.168
	0.8	0.097	0.126	0.154	0.193	0.238
	1.0	0.105	0.145	0.188	0.246	0.308
	1.2	0.111	0.165	0.220	0.299	0.375
	1.4	0.118	0.174	0.254	0.351	0.434
60	0.6	0.040	0.050	0.060	0.074	0.096
	0.8	0.043	0.055	0.079	0.103	0.153
	1.0	0.048	0.061	0.098	0.148	0.225
	1.2	0.052	0.067	0.121	0.201	0.304
	1.4	0.056	0.075	0.143	0.256	0.372

[a] For $+\phi$ values and for $-\phi$ values such that $-(-\phi)$ is less than ϕ (See Fig. A-1)

high transmittance. As an approximation the ground can be treated as a quarter-spherical source of uniform intensity. The transmittance of the shade for radiation from this source is computed in the same fashion except that negative profile angle values are used. Fig. A-4 gives some calculated values of A_d and T_d for diffuse reflecting slats for both above and below the horizon sources.

PART IV—TRANSMITTANCE AND ABSORPTANCE WHEN α VARIES WITH WAVELENGTH

In the usual case the slat surface absorptance for solar radiation varies with wavelength (see Fig. 3 of Reference 1 of the paper). For ordinary calculations, it is usually sufficiently accurate to use an average value α_{ave} of α, such that

$$\alpha_{ave} = \sum \alpha\lambda\, I\lambda \,/ \sum I\lambda \quad \ldots\ldots\ldots\ldots \text{(A-14)}$$

where

$\alpha\lambda$, $I\lambda$ equals the product of absorptance at a given wavelength, λ, and of $I\lambda$, the intensity of radiation at this wavelength. The summation is carried out over the

TABLE A-3—CONFIGURATION FACTORS F_1'[a]

SLAT ANGLE, ϕ DEG	W/S	m/W = FRACTION OF SLAT WIDTH IRRADIATED				
		0	¼	½	¾	1
0	0.6	0.243	0.269	0.296	0.328	0.362
	0.8	0.186	0.212	0.244	0.282	0.324
	1.0	0.146	0.174	0.202	0.243	0.292
	1.2	0.116	0.138	0.168	0.211	0.264
	1.4	0.034	0.114	0.142	0.184	0.242
15	0.6	0.332	0.368	0.404	0.442	0.480
	0.8	0.248	0.291	0.336	0.383	0.433
	1.0	0.184	0.231	0.278	0.332	0.391
	1.2	0.141	0.185	0.229	0.287	0.334
	1.4	0.097	0.148	0.199	0.247	0.323
30	0.6	0.442	0.485	0.528	0.572	0.607
	0.8	0.330	0.387	0.444	0.500	0.552
	1.0	0.240	0.302	0.364	0.434	0.500
	1.2	0.172	0.234	0.306	0.380	0.457
	1.4	0.139	0.194	0.249	0.328	0.414
45	0.6	0.580	0.624	0.668	0.709	0.737
	0.8	0.426	0.505	0.576	0.631	0.679
	1.0	0.300	0.386	0.472	0.551	0.617
	1.2	0.191	0.284	0.377	0.475	0.557
	1.4	0.117	0.207	0.297	0.405	0.503
60	0.6	0.749	0.782	0.815	0.842	0.863
	0.8	0.585	0.655	0.725	0.774	0.811
	1.0	0.381	0.492	0.603	0.684	0.742
	1.2	0.194	0.332	0.470	0.584	0.666
	1.4	0.098	0.229	0.360	0.473	0.600

[a] For negative values of ϕ such that $-(-\phi)$ is greater than ψ

spectral region of interest. α_{ave} may also be found from experimental determinations in sunlight.

For more exact calculations the fraction of the incident direct beam absorbed, A_D, and transmitted, T_D, is given by the following equations:

$$A_D = \sum A_{D,\lambda} I_\lambda \Big/ \sum I_\lambda \quad \text{(A-15)}$$

$$T_D = \sum T_{D,\lambda} I_\lambda \Big/ \sum I_\lambda \quad \text{(A-16)}$$

where

$A_{D,\lambda}$ and $T_{D,\lambda}$ are the absorptance and transmittance values for wavelength, λ, and depend upon the value of α for that wavelength.

It will be apparent that the energy distribution of the radiation which has passed through the sunshade will differ from that of the incident beam. This may be important if the shade is used with heat absorbing glass on the indoor side.

PART V—EFFECTIVE EMISSIVITY

The absorptance of low temperature radiation from a hemispherical black body radiation source is computed exactly as described in Part III of this appendix. These

TABLE A-4—CONFIGURATION FACTORS F_2'[a]

SLAT ANGLE, ϕ DEG	W/S	m/W = FRACTION OF SLAT WIDTH IRRADIATED				
		0	¼	½	¾	1
0	0.6	0.260	0.269	0.277	0.280	0.277
	0.8	0.314	0.337	0.352	0.356	0.352
	1.0	0.352	0.387	0.414	0.423	0.414
	1.2	0.384	0.435	0.469	0.482	0.469
	1.4	0.404	0.472	0.515	0.530	0.515
15	0.6	0.322	0.296	0.270	0.279	0.265
	0.8	0.386	0.382	0.378	0.361	0.339
	1.0	0.454	0.453	0.452	0.433	0.403
	1.2	0.503	0.509	0.515	0.495	0.459
	1.4	0.543	0.556	0.569	0.549	0.506
30	0.6	0.316	0.294	0.270	0.234	0.226
	0.8	0.421	0.399	0.357	0.334	0.300
	1.0	0.525	0.491	0.457	0.413	0.366
	1.2	0.598	0.567	0.536	0.485	0.427
	1.4	0.631	0.613	0.595	0.542	0.475
45	0.6	0.281	0.249	0.217	0.192	0.168
	0.8	0.422	0.372	0.322	0.275	0.238
	1.0	0.561	0.495	0.429	0.363	0.308
	1.2	0.677	0.601	0.529	0.447	0.375
	1.4	0.752	0.683	0.614	0.521	0.434
60	0.6	0.187	0.160	0.133	0.110	0.096
	0.8	0.378	0.302	0.226	0.183	0.153
	1.0	0.556	0.454	0.352	0.278	0.225
	1.2	0.750	0.615	0.489	0.381	0.304
	1.4	0.840	0.721	0.602	0.472	0.372

[a] For negative values of ϕ such that $-(-\phi)$ is greater than ϕ

calculations are carried out for both positive and negative profile angles and the results averaged. The slat surface absorptance, α, to be used is the emissivity of the slat surface. According to Kirchoff's law, the shade absorptance so determined is equal to the emissivity of the slat shade. The shade can be regarded as approximating a plane diffuse radiation source in computations which employ the usual radiation shape factors. The distribution of radiation from a plane such as a window parallel to the shade or from the total room enclosure closely approximates the distribution from a hemispherical source, and exchanges between a shade and these surfaces can be computed with good accuracy. Calculations of exchanges between a shade and the ceiling of a room, for example, are less accurately made.

Because a slat shade is not opaque to radiation, exchanges can take place between two surfaces separated by the shade. For example, a window provided with an indoor shade exchanges radiation through the shade with the indoor environment. The transmittance of the shade for low temperature radiation is therefore needed and is computed as outlined in Part III. In this case also calculations should be restricted to exchanges in which the directional characteristics of the shade can be averaged.

TABLE A-5—CONFIGURATION FACTORS F_3, F_3', AND F_4[a]

SLAT ANGLE, ϕ DEG	W/S	F_3	F_3'	F_4
0	0.6	0.361	0.361	0.278
	0.8	0.324	0.324	0.352
	1.0	0.292	0.292	0.416
	1.2	0.264	0.264	0.472
	1.4	0.241	0.241	0.518
15	0.6	0.471	0.261	0.268
	0.8	0.419	0.237	0.344
	1.0	0.374	0.215	0.411
	1.2	0.335	0.197	0.468
	1.4	0.301	0.180	0.519
30	0.6	0.588	0.175	0.237
	0.8	0.524	0.159	0.317
	1.0	0.460	0.147	0.393
	1.2	0.403	0.136	0.461
	1.4	0.363	0.124	0.513
45	0.6	0.711	0.102	0.187
	0.8	0.632	0.093	0.275
	1.0	0.547	0.087	0.356
	1.2	0.469	0.081	0.450
	1.4	0.402	0.076	0.522
60	0.6	0.839	0.047	0.114
	0.8	0.751	0.041	0.208
	1.0	0.636	0.042	0.322
	1.2	0.515	0.040	0.445
	1.4	0.424	0.037	0.539

[a] For all values of ϕ

PART VI—TWO-COLOR SLATS

Equations A-7, A-8, A-9 and A-13 can be modified to accommodate the situation where the opposite surfaces of the slat have different colors or absorptances. The absorptance of the slat surface irradiated by the radiant beam will be designated by α and that of the opposite surface by α'. With slats set in their normal position, that is, to exclude the direct sun, α would be the absorptance of the upper surface and α' that of the lower surface. If ground reflected radiation is being considered, this would be reversed. The equations are:

$$\text{Component } X = (1 - L)\left[F_1\,(1 - \alpha)\right] \quad \text{(A-17)}$$

$$\text{Component } Y = (1 - L)\left[\frac{F_2\,F_3\,(1 - \alpha)\,(1 - \alpha')}{1 - F_4^2\,(1 - \alpha)\,(1 - \alpha')}\right] \quad \text{(A-18)}$$

$$\text{Component } Z = (1 - L)\left[\frac{F_2\,(1 - \alpha)\,(1 - \alpha')}{1 - F_4^2\,(1 - \alpha)\,(1 - \alpha')}\right]\left[(F_3'\,F_4\,(1 - \alpha)\right] \quad \text{(A-19)}$$

$$\text{Absorptance } A_D = (1 - L)\left[\alpha + \frac{(1 - \alpha)\,F_2\,(\alpha\,(1 - \alpha')\,F_4 + \alpha')}{1 - (1 - \alpha)\,(1 - \alpha')\,F_4^2}\right] \quad \text{(A-20)}$$

TABLE A-6—VALUES OF THE CONFIGURATION FACTOR C_1[a]

SLAT ANGLE, ψ DEG	W/S	SLAT SURFACE ABSORPTANCE, α					
		0	0.2	0.4	0.6	0.8	1.0
0	0.6	0.50	0.38	0.26	0.16	0.08	0
	0.8	0.50	0.36	0.25	0.15	0.07	
	1.0	0.50	0.35	0.23	0.14	0.06	
	1.2	0.50	0.34	0.22	0.13	0.06	
	1.4	0.50	0.33	0.21	0.12	0.05	
15	0.6	0.58	0.44	0.32	0.20	0.10	0
	0.8	0.57	0.42	0.29	0.18	0.09	
	1.0	0.56	0.40	0.27	0.17	0.08	
	1.2	0.55	0.38	0.25	0.15	0.07	
	1.4	0.54	0.36	0.24	0.14	0.06	
30	0.6	0.67	0.52	0.38	0.24	0.12	0
	0.8	0.64	0.48	0.34	0.22	0.11	
	.0	0.61	0.45	0.32	0.20	0.10	
	1.2	0.59	0.42	0.29	0.18	0.08	
	1.4	0.58	0.40	0.27	0.16	0.08	
45	0.6	0.76	0.59	0.44	0.29	0.14	0
	0.8	0.71	0.55	0.40	0.26	0.13	
	1.0	0.67	0.50	0.36	0.23	0.11	
	1.2	0.63	0.46	0.32	0.20	0.10	
	1.4	0.61	0.42	0.28	0.18	0.08	
60	0.6	0.86	0.68	0.51	0.34	0.17	0
	0.8	0.79	0.62	0.46	0.30	0.15	
	1.0	0.73	0.55	0.40	0.26	0.13	
	1.2	0.66	0.49	0.34	0.22	0.11	
	1.4	0.63	0.43	0.29	0.18	0.08	

[a] For $+\phi$ values and for $-\phi$ values such that $-(-\phi)$ is less than ψ

TABLE A-7—VALUES OF THE PROFILE ANGLE ϕ IN TERMS OF ALTITUDE AND AZIMUTH

(For vertical slat type sun shades with slats horizontal)

ALTITUDE	AZIMUTH								
	5	15	25	35	45	55	65	75	85
5	5.00	5.18	5.62	6.07	7.07	8.58	11.8	18.8	45.2
15	15.1	15.5	16.5	18.1	20.8	25.0	32.4	46.0	72.0
25	25.1	25.8	27.2	29.6	33.4	39.1	47.8	61.0	79.4
35	35.1	35.9	37.7	40.5	44.7	50.7	58.9	69.7	82.9
45	45.1	46.0	47.8	50.7	54.7	60.2	67.1	75.5	85.0
55	55.1	55.9	57.6	60.2	63.7	68.1	74.0	79.7	86.5
65	65.1	65.8	67 1	69.1	71.8	75.0	78.9	83.1	87.7
75	75.1	75.5	76.4	77.6	79.3	81.3	83.5	86.0	88.7
85	85.0	85.2	85.5	85.9	86.5	87.1	87.9	88.7	89.6

TABLE A-8—VALUES OF THE CONFIGURATION FACTOR C_1'[a]

SLAT ANGLE, ψ DEG	W/S	SLAT SURFACE ABSORPTANCE, α					
		0	0.2	0.4	0.6	0.8	1.0
0	0.6	0.50	0.38	0.26	0.16	0.08	0
	0.8	0.50	0.36	0.25	0.15	0.07	
	1.0	0.50	0.35	0.23	0.14	0.06	
	1.2	0.50	0.34	0.22	0.13	0.06	
	1.4	0.50	0.33	0.21	0.12	0.05	
15	0.6	0.42	0.30	0.21	0.13	0.06	0
	0.8	0.43	0.30	0.20	0.12	0.05	
	1.0	0.44	0.30	0.20	0.11	0.05	
	1.2	0.45	0.30	0.19	0.11	0.05	
	1.4	0.46	0.30	0.18	0.10	0.04	
30	0.6	0.33	0.24	0.16	0.09	0.04	0
	0.8	0.36	0.25	0.16	0.09	0.04	
	1.0	0.39	0.26	0.16	0.09	0.04	
	1.2	0.41	0.26	0.16	0.09	0.04	
	1.4	0.42	0.26	0.16	0.08	0.03	
45	0.6	0.24	0.17	0.11	0.06	0.03	0
	0.8	0.29	0.20	0.12	0.07	0.03	
	1.0	0.32	0.21	0.13	0.07	0.03	
	1.2	0.37	0.23	0.13	0.07	0.03	
	1.4	0.39	0.24	0.13	0.07	0.02	
60	0.6	0.15	0.10	0.06	0.03	0.01	0
	0.8	0.21	0.14	0.08	0.04	0.01	
	1.0	0.28	0.18	0.10	0.05	0.02	
	1.2	0.34	0.20	0.12	0.06	0.02	
	1.4	0.37	0.21	0.12	0.06	0.02	

[a] For $-\phi$ values such that $-(-\phi)$ is greater than ψ

TABLE A-9—SPHERICAL AREA FACTORS TO USE WITH TABLE A-8[a]

ALTITUDE	AZIMUTH								
	5	15	25	35	45	55	65	75	85
5	0.00953	0.00925	0.00868	0.00784	0.00677	0.00549	0.00405	0.00247	0.00083
15	0.00897	0.00870	0.00816	0.00738	0.00637	0.00516	0.00381	0.00233	0.00078
25	0.00791	0.00766	0.00719	0.00649	0.00561	0.00455	0.00335	0.00205	0.00069
35	0.00646	0.00626	0.00587	0.00531	0.00458	0.00372	0.00274	0.00167	0.00056
45	0.00482	0.00467	0.00438	0.00396	0.00342	0.00277	0.00204	0.00125	0.00042
55	0.00317	0.00308	0.00289	0.00261	0.00225	0.00183	0.00134	0.00082	0.00027
65	0.00173	0.00168	0.00158	0.00142	0.00123	0.00099	0.00073	0.00044	0.00014
75	0.00066	0.00064	0.00060	0.00054	0.00047	0.00038	0.00028	0.00017	0.00006
85	0.00010	0.00009	0.00009	0.00008	0.00007	0.00005	0.00004	0.00003	0.00001

[a] From Reference A-3.

PART VII—EXAMPLES IN THE USE OF THE EQUATIONS

Example 1: Find the transmittance and absorptance of a slat-type sunshade for the following conditions:

$$W/S = 1.2,\ \psi = 30°,\ \alpha = 0.4,\ D/S = \text{zero, profile angle } \phi = 50°$$

Solution: From Equation A-6:

$$L = 1 - W/S\,\frac{\sin(\phi + \psi)}{\cos\phi} = 1 - \frac{1.2 \sin(50 + 30)}{\cos 50} = 1 - \frac{1.2 \times 0.985}{0.643} = -0.84$$

$$\frac{m}{W} = 1/(1 - L) = 1/(1 + 0.84) = 0.543$$

L is negative and therefore there is no straight-through component. L is assigned a value of zero in the equations for the reflected through components.
Reflected-through Component X (Equation A-7):
From Table A-1 by interpolation, $F_1 = 0.079$

$$X = (1 - 0)\left[0.079\,(1 - 0.4)\right] = 0.0474$$

Reflected-through Component Y (Equation A-8):
From Table A-2 by interpolation, $F_2 = 0.329$
From Table A-5, $F_3 = 0.403$ and $F_4 = 0.461$

$$Y = \left[1 - 0\right]\left[\frac{0.329 \times 0.403\,(1 - 0.4)^2}{1 - 0.461^2\,(1 - 0.4)^2}\right] = 0.0517$$

Reflected-through Component Z (Equation A-9):
From Table A-5, $F_3' = 0.136$

$$Z = \left[1 - 0\right]\left[\frac{0.329\,(1 - 0.4)^2}{1 - 0.461^2\,(1 - 0.4)^2}\right]\left[0.136 \times 0.461\,(1 - 0.4)\right] = 0.0048$$

Therefore:

$$T_D = 0 + 0.0474 + 0.0517 + 0.0048 = 0.104$$

As a check:

$$\text{Equation A-10} = \text{Equation A-8} + \text{Equation A-9}$$

From Table A-6, $C_1 = 0.29$ and Equation 10 evaluated is:

$$\left[1 - 0\right]\left[0.329 \times 0.29\,(1 - 0.4)\right] = 0.0572$$

$$\text{Equation A-8} + \text{Equation A-9} = 0.0517 + 0.0048 = 0.0565$$

which is within slide rule accuracy.

Note that $\frac{0.0474 + 0.0048}{0.104} = 0.50$. Therefore, half of the transmitted energy leaves the upper surfaces of the slats and moves in an upward direction, while the other half moves downward. The shade absorptance is, from Equation A-13,

$$A_D = 0.4\,(1 - 0)\left[1 + \frac{0.329\,(1 - 0.4)}{1 - 0.461\,(1 - 0.4)}\right] = 0.509$$

Example 2: Find the transmittance and absorptance of diffuse reflecting slats with the following conditions:

$$W/S = 1.0,\ \phi = 50°,\ \psi = 17°,\ \alpha = 0.4,\ D/S = 0.15$$

Solution: A correction for slat thickness-spacing ratio is required in this example. According to Part I the correction factor is $1/(1-P)$ where P from Equation A-12 is,

$$P = \frac{D/S \cos(\phi + \psi)}{\cos \phi} = \frac{0.15 \cos(50 + 17)}{\cos 50} = \frac{0.15 \times 0.391}{0.643} = 0.0913 \text{ and } 1/(1 - P) = 1.10$$

$$\therefore L = 1 - (1.10 \times 1.0) \frac{\sin(50 + 17)}{\cos 50} = 1 - \frac{1.10 \times 0.921}{0.643} = 1 - 1.575$$

$$L = -0.575$$

Therefore, the straight-through component is zero. From Equation A-11,

$$\frac{m}{W} = 1/(1 - L) = 1/[1 - (-0.575)] = 0.635$$

By interpolation from Tables A-1, A-2 and A-5, $F_1 = 0.136$, $F_2 = 0.388$, $F_3 = 0.365$, $F_3' = 0.197$, and $F_4 = 0.437$. This interpolation is carried out for (W/S) $[1/(1-P)]$, or 1.10 in this example. Then

$$X = 0.136 (1 - 0.4) = 0.0816$$

$$Y = \frac{0.388 \times 0.365 (1 - 0.4)^2}{1 - 0.437^2 (1 - 0.4)^2} = 0.0547$$

$$Z = \left[\frac{0.388 (1 - 0.4)^2}{1 - 0.437^2 (1 - 0.4)^2}\right] [0.197 \times 0.437 (1 - 0.4)] = 0.0077$$

$$X + Y + Z = 0.0816 + 0.0547 + 0.0077 = 0.144$$

Correcting T_D for slat thickness gives

$$T_D = (1 - P) 0.144 = (1 - 0.091) 0.144 = 0.131$$

The absorptance, A_D, with corrections for slat thickness is, from Equation A-13,

$$A_D = \alpha (1 - 0) \left[1 + \frac{0.388 (1 - 0.4)}{1 - 0.437 (1 - 0.4)}\right] (1 - P) + \alpha P$$
$$= 0.4 \times 1.315 (1 - 0.091) + 0.4 \times 0.091 = 0.515$$

In this example $\frac{0.0816 + 0.0077}{0.144} = 0.62$ of the transmitted radiation leaves the shade in a generally upward direction.

Example 3: Find the transmittance of a slat-type shade for diffuse radiation from an above-the-horizon quarter spherical source of uniform intensity. The shade has the following characteristics:

$$W/S = 1.2,\ D/S = 0.0,\ \psi = 30°,\ \alpha = 0.3$$

Solution: The results will be given and the procedure only indicated. The sum of the products of shade transmittance for each profile angle listed in Table A-7 and the corresponding spherical area factor of Table A-9 is (See Fig. 9 of Reference 3 of the paper for transmittance values):

$$\sum_{n=0}^{n=81} 0.439 \times 0.00953 + 0.283 \times 0.00897 + 0.188 \times 0.00791 + \cdots = 0.055$$

so that

$$T_d = 4 \times \sum_{n=0}^{n=81} = 4 \times 0.055 = 0.220$$

Example 4: Find the straight-through component, L, and m/W for a slat-type sunshade for the following conditions:

$$W/S = 1.0,\ \psi = 20°,\ \alpha = 0.3,\ D/S = \text{zero and } \phi = -40°.$$

Solution:

$$L = 1 + \frac{W}{S}\frac{\sin(\phi + \psi)}{\cos\phi} = 1 + 1.0\,\frac{\sin(-40° + 20°)}{\cos(-40°)} = 1 - \frac{\sin 20°}{\cos 40°}$$

$$= 1 - \frac{0.342}{0.766} = 1 - 0.447 = 0.553$$

$$\frac{m}{W} = 1/(1 - L) = 1/(1 - 0.553) = 1/0.447 = 2.24$$

In this case, m/W is assigned a value of 1.0. This is in accord with the fact that if part of the incident beam passes through the shade without striking the slats, then the entire width of the slat is irradiated by the radiant beam.

Example 5: Find m/W for the conditions of *Example 4* except that $\phi = -60°$.

Solution:

$$L = 1 + \frac{W}{S}\frac{\sin(-60° + 20°)}{\cos(-60°)} = 1 - 1.0 \times \frac{\sin 40°}{\cos 60°} = 1 - 1.0 \times \frac{0.643}{0.50}$$

$$= 1 - 1.286 = -0.286$$

This value is substituted directly in Equation A-12 so that,

$$\frac{m}{W} = 1/(1 - L) = 1/\left[1 - (-0.286)\right] = 1/1.286 = 0.778$$

Note, however, that according to rule 2(b) of Part I, the straight-through component is zero and that to evaluate the reflected-through components X, Y, and Z, L would be assigned a value of zero.

REFERENCES

A-1. A.S.H.V.E. RESEARCH REPORT No. 1460—An Analysis of the Effect of Uniformly Spaced Flat Opaque Slats, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 377).

A-2. Radiant Heating and Cooling, Part I, by C. O. Mackey, L. T. Wright, Jr., R. E. Clark and N. R. Gay (Cornell University Engineering Experiment Station, Bulletin No. 32, August 1943).

A-3. The Development of Prismatic Glass Block and the Daylighting Laboratory, by R. A. Boyd (Engineering Research Bulletin No. 32, University of Michigan, February 1951).

APPENDIX B

Transmittance of Glass-Shade Combinations

In the interest of completeness, equations for estimating the performance of shade-glass combinations previously given (Reference 3 of the paper) will be repeated in this appendix. Although an infinite series of reflections take place between shade and glass, the equations will take into account only one pair of reflections. Higher order reflections are usually of minor significance.

CASE I—FLAT GLASS BETWEEN THE SUN AND THE SHADE

For direct radiation the equations for shades with diffuse reflecting slat surfaces are: Transmittance of the combination:

$$T_{g,s,D} = \tau_D T_D + \tau_D R_D \rho_d T_d \quad \text{(B-1)}$$

Absorptance of the glass:

$$A_{g,D} = \alpha_D + \tau_D R_D \alpha_d \quad \text{(B-2)}$$

Absorptance of the shade:

$$A_{s,D} = \tau_D A_D + \tau_D R_D \rho_d A_d \quad \text{(B-3)}$$

Both T_d and A_d are averages of the values for positive and negative profile angles. Fig. A-4 gives values of T_d and A_d from which average values can be determined. It should be noted that A_D, T_D, and R_D depend upon the profile angle, ϕ, whereas τ_D and α_D depend upon the incident angle, Θ. In general ϕ and Θ are not the same. If the incident radiation is diffuse, all subscripts become lower case and the appropriate values for diffuse radiation are used.

CASE II—SHADE BETWEEN THE SUN AND FLAT GLASS

The following equations apply when the straight-through component L is zero. Transmittance of the combination:

$$T_{s,g,D} = T_D \tau_d + T_D \rho_d R_d \tau_d \quad \text{(B-4)}$$

Absorptance of the shade:

$$A_{s,D} = A_D + T_D \rho_d A_d \quad \text{(B-5)}$$

Absorptance of the glass:

$$A_{g,D} = T_D \alpha_d + T_D \rho_d R_d \alpha_d \quad \text{(B-6)}$$

R_d and A_d are averages of the values for positive and negative profile angles. Exact treatment of reflections when L is not zero is impracticable in most cases because the straight-through component is specularly reflected whereas the reflected-through components are diffusely reflected. However, the foregoing equations will give approximately correct results for this condition.

Example 1: Find the transmittance, $T_{g,s,d}$ and absorptances $A_{g,D}$ and $A_{s,D}$ of the shade in *Example 1*, Appendix A, when used indoors in combination with single common window glass. The incident and profile angles are 40 deg. In this case τ_d, α_d and ρ_d are respectively 0.79, 0.06, and 0.15 while τ_D and α_D are 0.86 and 0.06 respectively (See Table 12, Chapter 13, THE GUIDE 1953). $T_D = 0.115$ and $A_D = 0.515$ (Fig. 9, Reference 3 of the paper).

Solution: From Fig. A-4, the average of the T_d values for positive and negative profile angles is 0.41 and the average of the A_d values is 0.34.

$$R_D = 1 - A_D - T_D = 1 - 0.115 - 0.515 = 0.370$$

Therefore

$$T_{g,s,D} = 0.86 \times 0.115 + 0.86 \times 0.370 \times 0.15 \times 0.41 = 0.118$$
$$A_{g,D} = 0.06 + 0.86 \times 0.370 \times 0.06 = 0.079$$
$$A_{s,D} = 0.86 \times 0.515 + 0.86 \times 0.370 \times 0.15 \times 0.34 = 0.459$$

Example 2: Same as above except that the incident angle Θ is 60°, while the profile angle remains 40°.

α_D, α_d, τ_d and ρ_d are as before but $\tau_D = 0.79$

$$T_{g,s,D} = 0.79 \times 0.115 + 0.79 \times 0.370 \times 0.15 \times 0.41 = 0.109$$
$$A_{g,D} = 0.06 + 0.79 \times 0.370 \times 0.06 = 0.078$$
$$A_{s,D} = 0.79 \times 0.515 + 0.79 \times 0.370 \times 0.15 \times 0.34 = 0.422$$

APPENDIX C

The Shade Factor

The shade factor has been defined in the paper. In terms of calculable quantities, it can also be defined by the following:

$$\text{Shade factor} = \frac{T_{g,s,D}\, I_D + T_{g,s,d}\, I_d + n\left[(A_{g,s,D} - \alpha_D)\, I_D + (A_{g,s,d} - \alpha_d)\, I_d\right]}{\tau_{D,c}\, I_D + \tau_{d,c}\, I_d} \qquad \text{(C-1)}$$

where

n = an experimentally determined constant.
$\tau_{D,c}$ $\tau_{d,c}$ = transmittance for direct and diffuse solar radiation respectively; the subscript c refers to *common window glass.*
$A_{g,s,D}$, $A_{g,s,d}$ = $A_{g,D} + A_{s,D}$ and $A_{g,d} + A_{s,d}$ respectively.

Note that α_D and α_d in the numerator are values for the glass with which the shade is used. Reference 2 of the paper shows that for design purposes n may be taken as 0.75 for indoor shades and as 0.05 for outdoor shades for any type of single flat glass.

The subscripts as written are for indoor shades, but the same equation is used for outdoor shades, except that the order of the sugscripts g and s would be reversed.

Example 1: Find the shade factor for the shade and conditions of *Example 1* of Appendix B when I_D is 160 and I_d is 30 Btu per (hr) (sq ft), all from above the horizon.

Solution: With values of A_d and T_d for positive profile angles as given in Fig. A-4, $T_{z,s,d}$ and $A_{g,s,d}$ are found to be 0.158 and 0.476 respectively. Substitution in Equation C-1 gives a value of 0.560 for the shade factor.

Similar calculations for *Example 2* of Appendix B result in a shade factor of 0.545. It is useful to note that the transmittance and absorptance of glass-shade combinations, especially those in which the shade is indoors and also those outdoor shades in which the straight-through component is significant, are dependent upon both profile angle and incident angle. This was illustrated by *Examples 1* and *2* of Appendix B, although the difference was less than 10 percent. For most orientations the difference between τ_D for the incident and τ_D for an angle equal to the corresponding profile angle is usually small. Principal exceptions in the summer cooling period are early morning hours with southeast, and late afternoon with southwest orientation. (Compare the incident and profile angles of Fig. 6 of the paper.) However, the shade factor is little affected as has been illustrated. This fact makes the shade factor dependent on profile angle only for all practical purposes.

DISCUSSION

E. W. CONOVER, Detroit, Mich. (WRITTEN): We wish to commend the authors of than in other papers resulting from this project. On reading this paper a few questions have occurred to me.

It is stated that the surfaces of most painted slats in present use can be classed as diffuse reflectors. My observation is that most of them, particularly the metal ones, appear to have considerable specular reflection. Were the factors determined from tests on slats or calculated on the basis of diffuse surfaces?

In item 8 on page 407 it is stated that roughly ½ of the reflected-through component enters the room in a generally upward direction. This might be true if the slat surface is perfectly diffuse, but with some specular reflection it is believed that

it will be more than ½ and more nearly ⅔. What test data or computations influenced the authors to change this to ½?

In the section Design Assumptions it is stated that the test data showed that about 25 percent of the incident diffuse radiation came from the ground and 75 percent from the sky. Some surfaces such as snow, cement pavement sand, dry grass, etc., with the sun shining on them are brighter than the blue sky. I might ask how a 50-50 distribution between ground and sky would affect the design values of absorptance and transmittance.

R. W. MCKINLEY, Pittsburgh, Pa.: In the section Design Assumptions it is indicated that 25 percent of the diffuse radiation comes from the ground. Does this include both that radiation reflected from the ground and that re-radiated by the ground? Is this 25 percent portion applicable only to these test data, or do you propose that it is typical and may be used safely as a design figure?

In the case of a low south-facing window adjacent to a large high reflectance roof or ground area (sun altitude approximately 45 deg) I would expect that the diffuse component reflected might easily approach 75 percent of the direct solar radiation. Also, after the temperature of such a nearby radiating panel had been raised by continuous exposure for a period of several hours, it seems that the re-radiation to a window might be of significant magnitude.

Is this condition taken into consideration by these design data? If these factors are not included in the procedure proposed in the paper, how may they best be handled? The matter is mentioned in item 9 on page 407 and on page 412. Is this radiation considered in THE GUIDE 1953, Chapter 13, Table 6, used in Example 2?

W. A. DANIELSON, Raleigh, Tenn.: I wonder if the authors secured any information on the heat gain after the sun had passed the zenith with the slats pointing 45 deg downward, so that radiant heat would enter from heated ground surfaces. If they had the slats pointed 45 degrees upward or thereabouts, two results could take place. Radiation from objects outside would be eliminated. There might be some heat loss to the outer atmosphere. These might be more important than we realize.

AUTHORS' CLOSURE (G. V. Parmelee): Mr. Conover's comments with regard to the presentation of data and examples in the paper are appreciated. Such simplicity of presentation as may have been achieved is due largely to the advice received from the TAC on Heat Flow Through Glass as to content and type of data needed.

The statement that most painted slats in present use can be classed as diffuse reflectors is based upon the findings of spectral photometric tests as reported in the second paper on shading*. Several slat samples were tested and it was found that the specularly reflected component amounted to about 3 percent. Gonio-photometer test data were also used in estimating the magnitude of this component and from these it was noted that the specular component was found to be about 10 percent. These low values were contrary to the authors' first impressions upon examining the slats that they were specular reflectors because objects were sharply mirrored. It appears that one must conclude that the eye is not a good judge of quantity. The shade factors given in Table 1 of the paper are for completely diffuse reflecting surfaces and are based upon both tests of such surfaces and on calculations.

It is difficult to state exactly the fraction of the reflected-through component that enters a room in a generally upward direction. Calculations for a few typical conditions show that this fraction ranges from ½ to perhaps ⅔. It was felt that the rough figure of ½ was close enough for all except those who are interested in the detailed behavior of sun shades. Examples of the calculations required to determine this fraction are given in Appendix A to the paper. It does not necessarily follow

* A.S.H.V.E. RESEARCH REPORT No. 1474—The Shading of Sunlit Glass—An Experimental Study of Slat-Type Sun Shades, by G. V. Parmelee, W. W. Aubele and D. J. Vild (A.S.H.V.E. TRANSACTIONS, this volume, p. 221).

that if a slat surface is a specular reflector, the reflected-through radiation is thrown upward. For example, Fig. 1 of the first paper on shading† shows that under some conditions none of the reflected-through radiation is thrown upwards but is thrown downwards instead. Such a case occurs for a slat angle of 45 deg, a profile angle of 30 deg and $W/S = 1.2$. Each case needs to be individually examined, particularly for specular reflectors.

Mr. Conover asks what effect a change from a 25-75 percent distribution between ground reflected and sky diffuse radiation to a 50-50 percent distribution would have on values of absorptance and transmittance. For the specific condition of 45 deg slat angle and slat surface absorptance $\alpha = 0.4$, the 50-50 distribution increases the transmittance about 55 percent and decreases the absorptance about 15 percent. The data given in the three papers on shading have important applications to control of daylight, but these applications could not be considered in detail in any of the papers.

Mr. McKinley questions what part of the ground reflected radiation was assumed to be diffuse solar radiation and what part was re-radiation from the ground by virtue of its temperature. The 25 percent portion assigned to the ground and used as a basis of design applies only to solar radiation. It is doubtful that this figure is typical but in the absence of more complete information, it seemed wisest to use such information from our experimental work as we had. The figure we have used corresponds approximately to a ground reflectance of about 10 percent. The surroundings of the testing apparatus were dark on the whole, so that shade factors for windows which see light colored surfaces would be somewhat higher than those given in the paper. When a typical ground reflectance value can be stated and accompanying diffuse solar radiation values measured, shade factors can be modified, if necessary.

The effect of re-radiation of the ground and other surroundings which have been warmed by the sun is taken into account by the experimental observations of the convection and radiation gain from shaded windows. These observations were the basis, in part, of the shade factors.

Mr. McKinley's reference to *Example 2* and Table 6 of THE GUIDE 1953 apparently refers to Chapter 13 on Cooling Load. It is not clear what information his question seeks. Table 6 refers to the altitude and azimuth of the sun. Table 2 of the paper gives values of shadow line or profile angles for slat type sun shades installed in the usual manner for various orientations, and as such supplements Table 6 of THE GUIDE Chapter 13. The solar radiation values used in the design shade factors are those of Table 4 of Chapter 13 for clear atmospheres. The diffuse solar radiation values include both the sky and the ground components and are based upon observations in Cleveland over a considerable period of time. However, no attempt was made to determine the relative magnitudes of the two components. The estimate given in the paper in respect to this point was based upon comparing observed transmittance values for sun shades with computed values, and it was concluded that about 25 percent of the incident diffuse solar radiation was ground reflected radiation. It is true that if the ground has a high reflectance, the diffuse component of the total solar radiation which falls on a south-facing window in summer may be quite large. This is mainly because the direct component of solar radiation is relatively small because of the very high angle of incidence of the sun at all hours of the day.

General Danielson seeks information concerning heat gain after the sun has passed off the window, but while the ground is still warm. Experiments carried out to determine the convection and radiation gain from a shaded window facing away from the sun are the basis of the factors given for windows in the shade (see Table 1 of the paper). These factors include the effect of wind and low temperature radiation from the ground. No tests were made in which the slats were pointed 45 deg upward. It is likely that different results would have been obtained.

† A.S.H.V.E. RESEARCH REPORT No. 1460—The Shading of Sunlit Glass—An Analysis of the Effect of Uniformly Spaced Flat Opaque Slats, by G. V. Parmelee and W. W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 377).

1486

No. 1486

AUTOMATIC PERMEANCE MEASUREMENT BY THE PERMEOMETER

By F. A. JOY*, STATE COLLEGE, PA. AND A. W. SHERDON**, LANCASTER, PA.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with The Pennsylvania State College, State College, Pa.

THE CONTROL of water vapor migration is vital to the packaging engineer and to the architect whose designs also become packages. Moisture exclusion is the objective in the packaging of metal parts while moisture retention is important in products such as foods and tobacco. A vapor barrier, or properly a vapor resistor, restricts moisture transfer to a permissible rate which depends on the application. An excellent barrier is necessary for long term storage while a higher transfer rate is allowable in house construction where the barrier function is seasonal. Very high moisture transfer, on the other hand, is sometimes an objective. For any application, the choice of a vapor control membrane involves its *permeance*, a characteristic of the sheet or film that requires definite measurement.

The *permeance* of a sheet is the reciprocal of its resistance to vapor passage. It is the ratio of the moisture transmitted through a unit area, in unit time, to the vapor pressure difference, the conditions being known. The term *permeability*, often loosely used in this sense, correctly relates to a property of the substance and is the permeance of a unit thickness of the material. The unit of permeance generally used by the building industry is the *perm* which is one grain per (sq ft) (hr) (in. Hg difference of vapor pressure). These relations are expressed in the following formula:

$$\text{Permeance} = W/(ATΔp) \quad \ldots\ldots\ldots\ldots \quad (1)$$

* Professor of Engineering Research, The Pennsylvania State College. Member of A.S.H.V.E.

** Engineer, The Armstrong Cork Co. (Formerly Research Associate, The Pennsylvania State College.)

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

where

W = the weight of vapor, grains.
A = area, square feet.
T = time, hours.
Δp = vapor pressure difference, inches of mercury.

The measurement of permeance of a sheet material is generally accomplished by sealing a specimen to the top of a dish or cup which contains water or a desiccant, exposing it in a room or cabinet where the air conditions are carefully controlled, and weighing the assembly periodically to determine its rate of weight change. After a steady state has been established, the permeance can be calculated. This simple procedure has a number of variations but all involve weighing and therefore a considerable time may be necessary to obtain a weight change large enough to minimize the effects of scale insensitivity and of operational error.

To find a faster method was the original purpose of the project, started in 1950, in which the *Permeometer* was developed. Other objectives were soon added. These included:

1. A transfer of skill and accuracy from the operator to the machine.
2. A single observation of the measurement by the operator.
3. Elimination of humidity control from the operating space.
4. Elimination of precise temperature control and acceptance of the ambient temperature within the usual comfort range.
5. Easy standardization of the instrument.

METHODS INVESTIGATED

To accomplish these objectives, consideration was given to several possibilities, all of which eliminate the need for weighing. The methods considered were based on the following natural phenomena:

1. The change in the dielectric property of a desiccant with changing water content.
2. The reflection of light from a cold mirror on which water vapor is allowed to condense.
3. The vapor pressure drop across an orifice in the vapor path.

Since the convenience of an electric measurement was attractive, a device based on the first principle was constructed. It indicated the changing capacitance at radio frequency of a condenser whose dielectric was a thin bed of silica gel used as the water vapor sink. Observations, however, indicated that the quantity measured, in relation to the moisture added, was a function too insensitive for quick results. It was also doubtful whether a single observation of the instrument would be adequate in a test. Other problems encountered added to the difficulties of the method and its development was, therefore, suspended.

Utilizing the second principle, another device was built which promised some convenience in obtaining a low pressure by the use of ice. After a steady state was reached a small silver mirror could be moved into the cold zone where it would condense the vapor flow in a small area. Photometer measurement of reflected light was a feature of the design which appeared attractive. However,

the exacting control of heat transmission and temperatures that was needed presented obstacles and suggested that another approach should be investigated.

ORIFICE MEASUREMENT

The use of an orifice is the basic feature of the *Permeometer*. Orifice measurement of liquid or gas flow is accepted as a quick and dependable method but its use in measuring vapor diffusion has been somewhat neglected. Its utility was recognized by *The Institute of Paper Chemistry* as early as 1944 when a device providing either a fixed or manually adjustable vapor resistance was described.[1]

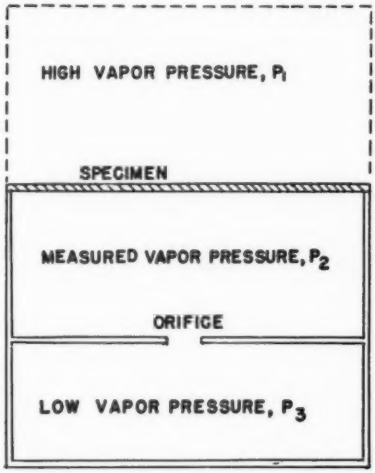


FIG. 1. ORIFICE MEASUREMENT OF VAPOR DIFFUSION

The study of vapor flow through holes, made by Yeaple[2] in 1948, indicated that the vapor transfer may result from air convection as well as vapor diffusion. Such behavior is expected where a difference of air density provides an unstable condition, and its avoidance is vital to the procedure considered. As dry air is more dense than humid air the measuring orifice should be located in a horizontal plane with dry air below and humid air above it. The reversed conditions in the earlier device of *The Institute of Paper Chemistry* may have caused some irregularity in results although a "fairly accurate correlation" with gravimetric measurements of vapor transfer was reported. Unfortunately the gravimetric determinations were made under different exposure conditions, a fact that obscures the true accuracy of the method.

The essential requirements for orifice measurement with vapor flow downward are shown in Fig. 1. The orifice could be placed above the specimen instead of below, though the down-stream location is preferred for operational

[1] Exponent numerals refer to References.

convenience in the apparatus developed, and for greater versatility. The high vapor pressure, P_1, may be the ambient pressure, if it is controlled; but generation of vapor within an enclosure as indicated is readily accomplished. The low pressure, P_3, should be essentially constant and is maintained conveniently by a desiccant.

In Fig. 1, the fixed orifice restricts vapor diffusion and the vapor pressure above it reaches P_2 when a steady state is established. If P_2 is measured and

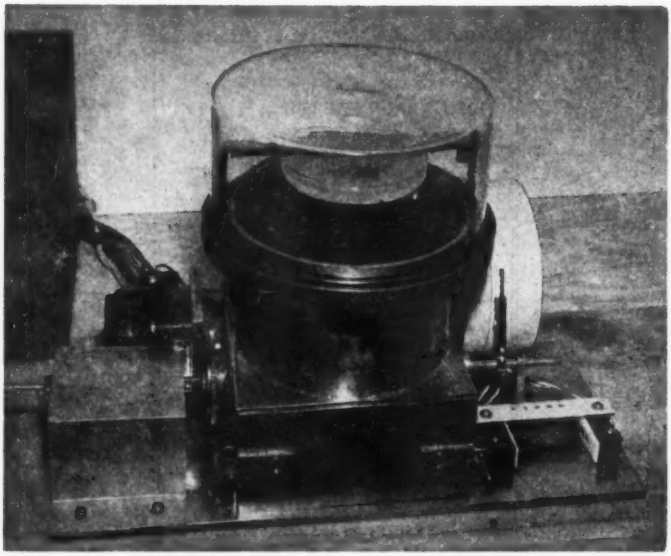


FIG. 2. THE PERMEOMETER

the orifice calibrated, the determination of vapor flow is simple and the permeance becomes:

$$\text{Permeance} = f(P_2 - P_3)/A(P_1 - P_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

f = calibration factor for orifice.

While the simple device is adequate in theory, it has certain objections. Accurate measurement of P_2, which will vary over a wide range for different specimens, becomes especially important at either limit. Such accuracy presents considerable problems and the widely different exposure for different specimens is also undesirable. Such problems are readily avoided by choosing a suitable orifice for each specimen or, more conveniently, providing an adjustable orifice.

AUTOMATIC ADJUSTMENT

Automatic adjustment of the orifice is a feature of the *Permeometer*. Essentially it holds a chosen vapor pressure, P_2, in the intermediate chamber. As there must be a different orifice size for each vapor flow rate and as each vapor flow rate results from a specific permeance of the specimen, it follows that each permeance requires a specific orifice opening. In short, the orifice opening is a direct function of the specimen permeance. When a steady state exists, it is necessary only to observe the orifice opening, converting its position to perms

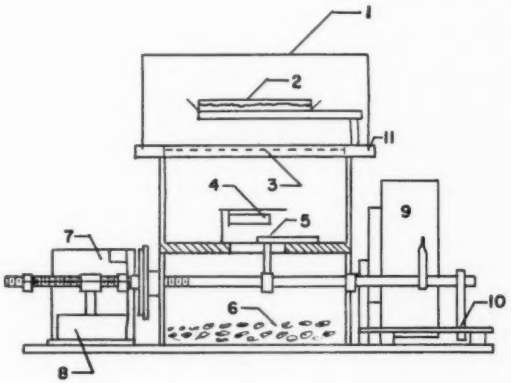


FIG. 3. THE PERMEOMETER ASSEMBLY (ELEVATION)

1. Glass cover
2. Salt solution, vapor source
3. Specimen in removable ring
4. Electric hygrometer, control element
5. Adjustable orifice and slide valve
6. Desiccant, vapor sink
7. Two phase balancing motor
8. Arresting potentiometer
9. Strip chart recorder
10. Valve position scale and pointer
11. Annular oil seal
12. Amplifier (not shown)

by reference to a calibration. Indeed, if desired, it is possible to graduate the indicating scale of the instrument in perms.

The device constructed after some preliminary designs, is shown in the photograph, Fig. 2, and its assembly is indicated in Fig. 3. This design provides the essential elements shown in Fig. 1 with the addition of automatic orifice adjustment and other features necessary to more complete self containment and ease of manipulation.

At the top, the vapor source is enclosed in a removable glass canopy. It is a saturated solution of sodium nitrite, chosen to provide a relative humidity in the 60 percent range. This is higher than the accepted level for *dry cup* tests but the difference is in a range where no major shift of permeance is ordinarily observed and, consequently, the exposure may be considered comparable to the standard. The glass dish holding the solution is placed on a swinging arm which may be moved to one side when the specimen is changed.

The specimen is mounted within a 6 in. diameter ring using the customary waxing technique for sealing the edge. The ring may be duplicated to allow a quick change of specimens and, in position, is sealed to the body of the apparatus. An oil seal has been used for this ring and also for the glass canopy as shown in Fig. 3. This type is very efficient and permits a quick change. However, the dripping of oil at the time of removal is an objection correctible by a wiping device not shown in the illustration. A synthetic rubber seal may also be adequate, a possibility to be investigated.

In the base of the apparatus, the low vapor-pressure chamber has a door at the front to admit a tray containing desiccant (silica gel). Vapor enters the top surface of the desiccant bed.

The measuring or control chamber is separated from the bottom compartment by a brass plate which has a 2 in. diameter orifice. Orifice adjustment is accomplished by a slide valve (Fig. 3, part 5) whose active contour is designed to provide large valve movement per unit area near the closed position but has a steadily increasing rate of area change as the valve is opened. This insures good sensitivity at any orifice opening. The valve is actuated by a shaft which moves axially below the orifice and passes through opposite sides of the desiccant chamber. This through-movement eliminates any pumping of air. Packing of the shaft at its bearings is not important since any stray vapor entrance is immediately absorbed in the desiccant without entry into the measuring path.

Automatic adjustment of the valve is effected by a two-phase balancing motor which is connected through suitable gears to the threaded shaft and is controlled by the vapor pressure in the measuring chamber. The sensing element is a commercial electric hygrometer duplicating in principle the sensing method used by *The Institute of Paper Chemistry*.[1] This device is very responsive to relative humidity and is able to control the vapor pressure at a chosen value if the temperature is constant. (While constancy of temperature is presumed, variations are later considered.) The vapor pressure to be maintained in the measuring chamber has been arbitrarily chosen, about midway between the high and low vapor pressures. The corresponding relative humidity, 30 percent in the basic conditions, is shown in Fig. 4 at valve position 11. (Position is indicated on a uniform scale graduated with 28 units per inch.)

Fig. 4 shows that the top and bottom exposures of the specimen are not the same for all specimens. The vapor pressure above the specimen naturally diminishes as the rate of evaporation from the salt solution is increased. The vapor pressure below the specimen, increasing with the vapor flow, is so arranged as a matter of control stability which is later considered. These pressure changes however are not an obstacle to correct measurement. For any given permeance, or valve position which is a function of permeance, both exposures are fixed and the vapor pressures are definite. The altered pressures that are provided for another specimen of different permeance are fully taken into account in the calibration. The exposures, of course, are different from those provided in standard dry-cup tests and the *Permeometer* measurement is not expected to match the dry-cup permeance. It is, however, expected to be similar since permeance is a ratio which relates the vapor flow to the existing vapor pressure difference (Equation 1) and since experimental evidence indicates no wide change in permeance within the range of relative humidity under consideration.[3,4]

The electric control circuits that were developed for this instrument are shown in Fig. 5. Basically, a Wheatstone bridge is used in which one arm is the sensitive resistance of the electric hygrometer. The bridge output is amplified and supplied to the two-phase balancing motor which moves the valve to effect control.

Control of vapor diffusion, however, is a different matter from the control of a mass flow of water or steam by a valve. Diffusion response is necessarily slow, in fact it is so delayed that the valve may be moved a great deal with

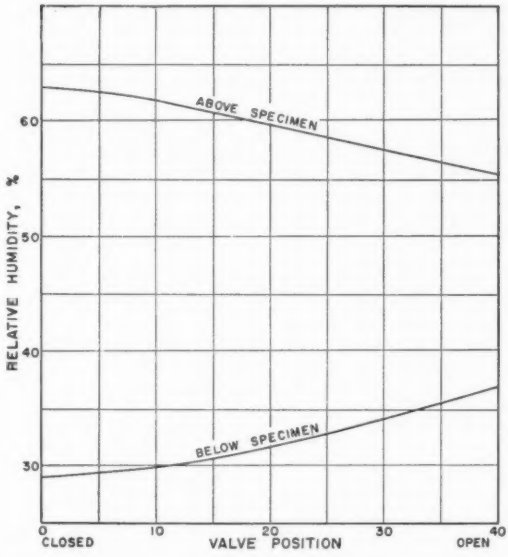


FIG. 4. TEST CONDITIONS AT 80 F (SODIUM NITRITE VAPOR SOURCE)

no immediately sensible change in the relative humidity above it. As a consequence, the basic circuit cannot function satisfactorily. An *open* signal from the electric hygrometer would cause the valve to open wide. This, of course, would soon be followed by a *close* signal which would continue until complete closure would result. These oscillations would continue indefinitely and are caused not so much by insensitivity of the hygrometer element, as by the resistance of the air path to vapor diffusion and, to a minor extent, by the adsorption of vapor in the control chamber. The apparent effect is inertia of the vapor pressure.

A simple means of overcoming this difficulty is the *cancelling circuit* shown in Fig. 5. The essential element is the potentiometer (part 1) actuated by the valve movement as indicated in Fig. 3. A signal from the electric hygrometer

that the relative humidity is too high opens the valve but the valve also moves the potentiometer, immediately restoring the balance and stopping the movement. This action effectively prevents hunting. It results, of course, in a controlled relative humidity that is higher at the open limit of the valve movement as shown in the upward sloping line in Fig. 4. The degree of slope is chosen for good regulation.

STRIP CHART RECORD

When a specimen is placed in the *Permeometer*, the vapor pressure in the control chamber, resulting from the entrance of ambient air, is likely to be above its control range and the resulting signal causes the valve to open wide. The excessive relative humidity drops to 37 percent immediately and the valve starts to close. Its movement, however, is not likely to be fast as the relative

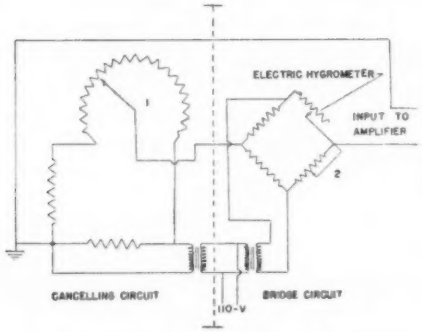


FIG. 5. PRIMARY ACTUATING CIRCUIT

humidity continues to drop and the moisture which may be stored in the lower face of the specimen (if it is hygroscopic) is withdrawn. In time, a steady state is reached where the vapor supply to the control chamber is equal to the loss therefrom through the orifice. Thereafter, the valve position remains essentially unchanged, except for slight oscillations associated with electrical and other disturbances.

It is important that a record of the valve movement during the test be available to the observer so that he may know whether a constant position has been reached or the slow change is still continuing. This record is supplied by a strip chart carried on a rotating drum that is powered by an electric clock mechanism providing one revolution per day. The pen is carried by the shaft extension as also is the valve position pointer. The strip chart may be ruled in units of valve position (or of perms in a special case). Eight specimen charts are shown in Fig. 6. In each case, the test was continued several hours to show fully the behavior of the instrument, although the permeance was clearly indicated for some specimens in a much shorter time.

While the permeance of some papers may be measured in an hour, others require much longer. The weight of the stock, its hygroscopicity and previous

exposure are factors that determine the amount of moisture stored in the specimen at the start. The test exposures usually involve an addition of moisture in the top fibers and extraction from the bottom fibers and the time required for these changes depends on the permeability of the material as well as the amount of moisture to be moved. These differences are evident in Table 1 which shows the time required for tests on various papers, mostly used in

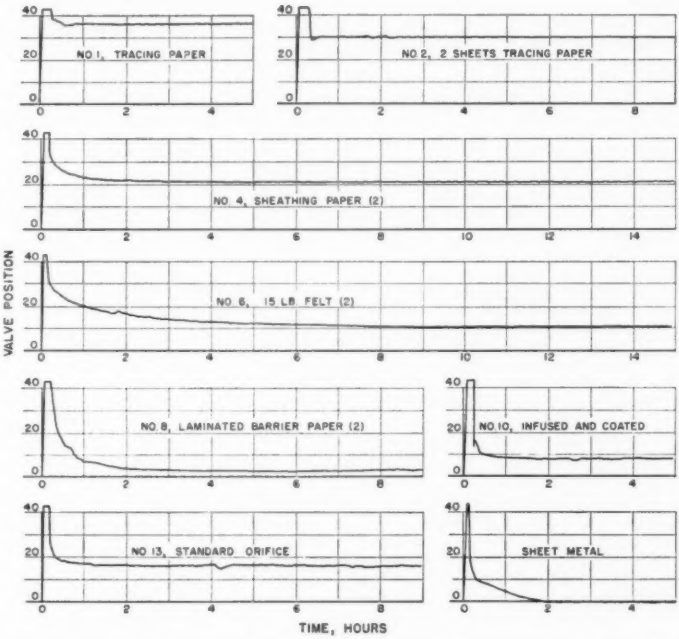

NO 1, TRACING PAPER
NO 2, 2 SHEETS TRACING PAPER
NO 4, SHEATHING PAPER (2)
NO 6, 15 LB FELT (2)
NO 8, LAMINATED BARRIER PAPER (2)
NO 10, INFUSED AND COATED
NO 13, STANDARD ORIFICE
SHEET METAL
VALVE POSITION
TIME, HOURS

FIG. 6. RECORDER STRIP CHARTS (SPECIMEN NUMBERS IN TABLE 1)

building, with the addition of other specimens suitable for the calibration of the instrument. The heaviest material shown, 15 lb felt, required 8 hrs for specimen 1, which had been in the laboratory for some time, whereas specimen 2, taken directly from an unheated storage (in October) required 13 hr. All other specimens had incidental laboratory exposure before they were tested. From these results, it appears that the permeance of building papers up to 15 lb weight can usually be measured in 8 hr, with 2 hr adequate for lighter papers.

No direct comparison of the *Permeometer* speed of measurement with that of a weighing method was attempted. In the latter case, the time is likely to be rather long since it is more a matter of efficient or convenient laboratory routine. It can be reduced, of course, by frequent and very exact weighing. In either

method, a steady-state must be first established with as much vapor leaving one side of the specimen as is entering the other side. The periods required for this preliminary may be similar, although each can be shorter in certain cases. In the *Permeometer*, however, the measurement is complete as soon as a steady state is attained, whereas the weighing procedure begins to be useful at this time. Thereafter, the weighing must be continued long enough to record a change suitably large in relation to the weighing tolerance. When it is considered that a one-perm barrier paper in a 30 sq cm dry cup test[5] at 73.4 F allows only 21 milligrams weight gain in a day, it is evident that a 10 percent tolerance of results requires careful handling and at least one day of the steady state condition. Better barriers require a proportionately longer time.

TABLE 1—TESTED PERMEANCE OF PAPERS (AT 80 F)

SPECIMEN NO.	DESCRIPTION	WEIGHT OF 100 SQ FT	PERMEOMETER			DRY CUP PERMEANCE PERMS
		POUNDS	Time**	Position	Permeance Perms	
1	Tracing Paper (100% rag)	1.4	½	36.6	11.0 *	—
2	same, 2 sheets		¾	29.5	7.1 *	—
3	Sheathing paper, asphalt infused kraft (1)	2.7	8	26.2	5.6 *	5.4
4	same, another specimen (2)		7	21.2	3.6 *	3.5
5	15 lb Felt, roll roofing (1)	14.1	8	10.2	0.81*	0.75
6	same, another specimen (2)		13	10.8	0.95	—
7	Laminated and reinforced barrier paper (1)	7.2	8	4.5	0.27*	—
8	same, another specimen (2)		7	2.7	0.17	0.19
9	Asphalt saturated and coated barrier paper	10.5	5	6.8	0.43	0.46
	Insulation back-up papers:					
10	(*a*) Asphalt infused and coated one side	6.1	2	8.3	0.59	0.58
11	(*b*) Waxed one side, asphalt side ribbed	3.3	2	9.3	0.72	0.66
12	Cellulose acetate	6.6	2	23.5	4.4 *	4.6
13	Standard orifice, 3/16 in. D (equivalent perms)		1	16.3	2.2 *	—

* Permeance obtained simultaneously by weight and used for *Permeometer* calibration curve, Fig. 7.
** Time (hours) required for the *Permeometer* measurement.

The preliminary time necessary to establish the steady state has not been critically examined for either method. It is significant, however, in the method described that the average of the top and bottom exposures of the specimen is about 46 percent relative humidity which may be near to laboratory conditions. When this is true, the small rise (17 percent RH, or less) above the specimen when it is placed under test is about balanced by the relative humidity drop under the specimen, and the resulting changes in hygroscopic moisture are minimized, with a saving in time expected.

The calibration of the *Permeometer*, basically at 80 F, is presented in Fig. 7 showing the relation between valve position and permeance of the specimen simultaneously determined by the gravimetric method. The calibrating procedure required several specimens in a wide range of permeance with the vapor transfer accurately weighed after the steady state had been established and was being normally maintained. The weight determination was made by substituting a special canopy with a pipe stem of some length rising above it as a diffusion seal. Through this pipe, a wire was hung from an analytical balance to support

the small glass dish containing the vapor source. Periodic weighings measured the rate of weight loss of the source which was the vapor transfer. Between weighings the dish was normally positioned by detaching the support wire from the balance and allowing a rubber plug on the wire to stopper the top end of the pipe stem. The vapor pressures above and below the specimen were determined by calibrated electric hygrometers and the permeance of each specimen was calculated by the formula, Equation 1. Eight specimens were used and the plot of each result is shown in Fig. 7 as a calibration curve at 80 F.

STANDARDIZATION

One of the specimens was an orifice 9/16 in. in diameter in a ⅛-in. plate. This may be taken as a standardizing device, since its vapor transmission rate under given conditions is not subject to change. By this standard, the performance of the device can be easily checked and immediately restored to calibration if any change has occurred. Such a change in the electric hygrometer may occur in time, and, indeed, another hygrometer element of different calibration may be substituted. In either case, standardizing is accomplished merely by adjusting the bridge circuit, Fig. 5. The potentiometer (part 2) is reset so that the valve automatically comes to rest at position 16.3 as recorded in Table 1, the temperature being 80 F. The validity of this simple procedure is provided in standard dependable elements: (1) vapor pressure of the source; (2) vapor pressure of the sink (with fresh desiccant); (3) exact duplication of the original vapor path. These conditions insure duplication of the original vapor pressure in the control chamber and the adjustment provides the correct response to that vapor pressure. Changes in other components of the primary actuating circuit are likely to be small and are corrected in the overall check described. Gross change, or failure of a component, is expected to be evident in the behavior. Changes in the amplifier also are similarly indicated.

In practice, erroneous results are avoided by: (1) periodic standardization; (2) maintenance of the water level at the source and (3) maintenance of fresh desiccant. Although the desiccant holding capacity is large, the dryness of the desiccant is assured only by a supply in good condition and frequent change. An additional feature, not shown in Fig. 3, is readily arranged, however, by placing a second appropriate electric hygrometer in the bottom chamber to provide a definite check of the desiccant condition.

Reproducibility of results is essential in any measuring device but failure to attain this goal has always been a problem in water vapor transfer measurements. Experience with this instrument has indicated reproducibility within 10 percent for the same specimen. The difference in two specimens of the same paper is another matter, which the instrument detects but cannot eliminate. Such differences are shown in Table 1.

Table 1 also present the results of dry cup measurements on eight specimens previously tested in the *Permeometer*. The dry cup required a slightly smaller diameter and, therefore, the specimens are not strictly duplicates but are considered essentially so. However, a comparison of these results is not merely a comparison of an automatic device with a weighing technique since the permeance of the specimen is not a fixed property. Actually, it is also a comparison of the specimen's permeance under one set of conditions as shown in Fig. 4

with its permeance under another set of conditions (50 and 0 percent RH) which is considered standard for some purposes. The *Permeometer* results range from 10 percent below to 9 percent above the dry cup data, with the largest percentage applying to a permeance of 0.19 perm where a ratio is hardly justified. In fact, this permeance is below the satisfactory range of the instrument which is 0.3 perm to 12 perms as adjusted. (A lower range of permeance could be obtained with the same instrument by adjusting the vapor pressure below the specimen, Fig. 4, to a lower range.) However, this agreement of the two methods is close. It is doubtful if such agreement will be found for all

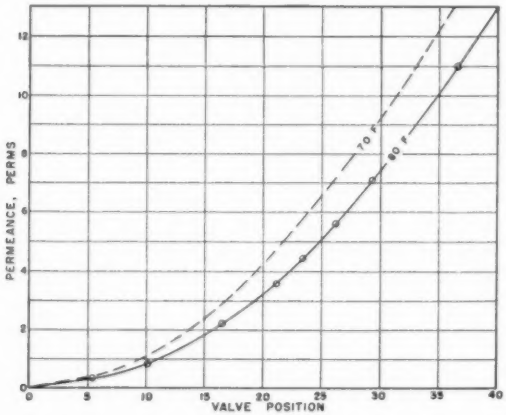


FIG. 7. PERMEOMETER CALIBRATION

materials, but if the exposures provided in the *Permeometer* are more like those encountered in service, its results are preferable for use.

The calibration curve, Fig. 7, shows the permeance of any membrane in the exposures actually provided and a clearly defined line is thus obtained. No scattering of test points is occasioned by the identity or characteristics, such as weight and hygroscopicity of the test specimen, but this effect would, of course, result from any failure of reproducibility by the device. An alternative calibration, or rather a correlation, could be considered, *i.e.*, plotting the *dry-cup permeance* against *valve position*, if this basis seemed desirable. This would occasion a considerable scattering of points due to the individualities of the test specimens and many more tests would be needed to obtain a representative correlation.

Operation of the *Permeometer* has been observed principally in a controlled temperature room at 80 F. Additional work at other temperatures is in progress, however, and a tentative calibration at 70 F is also shown in Fig. 7. Other curves at suitable intervals of temperature can be established. With this family of curves, it is evident that the *Permeometer* may be operated in any room having controlled temperature. It is also probable that an uncontrolled room, where temperature changes are moderate and slow, will suffice for ordinary

requirements where precision is not required. Observing a thermometer within the glass canopy and the valve position, the operator enters his observations in Fig. 7, where interpolation provides the correct permeance.

Another application also requires further investigation. This is a permeance measurement that is comparable to the standard *wet cup*. It is expected that the *Permeometer* will make this measurement with equal facility when pure water (or a high vapor-pressure solution) is substituted for the sodium nitrite solution. Another family of calibration curves will apply.

Although experience with this instrument is limited, it has been demonstrated that automatic orifice measurement of vapor diffusion is practicable and that the *Permeometer* provides a quick, convenient determination which heretofore has been considered an exacting task requiring laboratory facilities and considerable skill on the part of the personnel. Its range of measurement is adequate for a wide variety of building papers. Thicker materials may also be tested though the time saving will not be important. Its accuracy and the reproducibility of results require further study, though these virtues appear to be adequately assured for the purposes considered. Its application may be found in quality control and development needs or in small laboratories where a more extensive installation is not justified.

ACKNOWLEDGMENT

The initial development of the *Permeometer* was made possible by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in a research fellowship at The Pennsylvania State College and further advances were made under College sponsorship. The authors are indebted not only to the Society but to F. H. Yonker and P. L. Neyhart of the Engineering Staff and other associates, who contributed important advice and assistance.

REFERENCES

1. Instrumentation Studies LII. (Part V of Report No. 30 presented to the *American Paper and Pulp Association* on February 1, 1944, by *The Institute of Paper Chemistry*).
2. The Water Vapor Transmission Test Cell, by F. D. Yeaple (a thesis submitted in partial fulfillment of the requirements for the degree of Master of Science, June 1948, The Pennsylvania State College).
3. Water Vapor Transfer through Building Materials, by F. A. Joy, E. R. Queer, and R. E. Schreiner (Bulletin #61, Engineering Experiment Station, The Pennsylvania State College).
4. Permeability of Membranes to Water Vapor with Special Reference to Packaging Materials, by F. T. Carson (Miscellaneous Publication M 127, U. S. National Bureau of Standards).
5. *A.S.T.M.* Committee E-1 Proposed dry cup method.

DISCUSSION

R. R. SULLIVAN, Cloquet, Minn. (WRITTEN): The authors are to be commended for the development of the *Permeometer*. It is an instrument which deserves further study relative to its performance and to the utilization of its test values.

The objective of a rapid procedure for determination of permeance involving a single reading capable of being performed by a relatively non-skilled person is a

practical one. How well the *Permeometer* with its associated amplifier and cancelling circuit will fulfill this purpose over a period of time in the hands of non-skilled operators can only be found by trial. As pointed out in the paper, the results obtained are more suitable for control purposes or end use applications than for use in specifications calling for standard procedures such as those of the *Technical Association of the Pulp and Paper Industry* and *A.S.T.M.* since the conditions of test may be different for each material tested.

Certain questions arise concerning the instrument: (1) Would it be desirable to standardize at more than one point, that is have two standard plates bearing orifices of different size to determine more adequately whether the equipment has changed characteristics? (2) Even though the valve drive shaft extends through the desiccant chamber it is well above the desiccant, and is it not, therefore, desirable to have very good seals where it enters and leaves the chamber? (3) Some moisture vapor resistors contain volatile components or saturants which give rise to difficulty of measurement by the cup method. What is the performance of the *Permeometer* with such materials?

W. A. DANIELSON, Raleigh, Tenn.: I feel that vapor barriers are one of the most important developments in construction in recent years. I wonder if this instrument could be so arranged that it could be placed against the wall of an existing building in order to determine the amount of vapor coming through the wall.

AUTHORS' CLOSURE (F. A. Joy): Mr. Sullivan has made a very good suggestion. A second standard plate would be added insurance that the electric hygrometer was operating on its characteristic ratio of resistance *vs.* relative humidity. Experience shows that this is a very reliable relation, however. The only other change of the equipment affecting calibration would necessarily be in the cancelling potentiometer which is a simple and dependable element.

Seals for the valve shaft are considered unnecessary because of the large area of desiccant and because vapor diffusion does not occur in long restricted cracks. Another movement, air leakage, is considered desirable to insure atmospheric pressure. A capillary tube has been used between the source and sink chambers to insure equality of total pressures on the specimen.

The problems posed by evaporating specimens are annoying when weight is the basis of measurement. No experience with these materials in the permeometer is available but it appears that no error would result unless the electric hygrometer were affected by the gas released.

General Danielson's requirement—the measurement of vapor transfer in a field situation—is a recognized need. The principles of the permeometer, adapted to a vertical arrangement, could be used. However, quick results would not be expected and a large area for the test would be recommended.

1487

No. 1487

THERMAL PERFORMANCE OF FRAME WALLS

Part II—Air Spaces Blocked at Mid-Height

By G. O. HANDEGORD* AND N. B. HUTCHEON**, SASKATOON, CANADA

THE FIRST paper[1] presented on this subject dealt with the vertical variations in temperatures and heat flow rates resulting from convection in air spaces in frame walls. Some of the walls tested had insulation ideally applied while others in which insulation was installed to form two air spaces, had gaps left at the top and bottom. The air spaces were continuous over the height of the wall. The present paper deals with walls having horizontal blocking at mid-height, and is concerned, as was the first paper, with the deviations of actual heat flow and temperature patterns from those predicted by simple theory.

Blocking may be installed in frame walls to provide nailing girths and added rigidity. When used for these purposes it need not block off the air space completely. It is frequently required, however, as firestopping and to be effective, must present a barrier to vertical air movement. Walls with complete blocking may be expected to show two convection patterns over their height.

The effect of through framing members on overall wall conductance is usually calculated by one of two methods given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1953, Chapter 9. In the first of these, the wall is treated as a series of separate parallel heat flow paths only, and in the second as a combination of series and parallel paths. Neither of these methods provides a rational basis for calculation of surface temperature under the actual conditions of heat flow in the wall.

The horizontal surface temperature variation in frame walls will be largely influenced by the type and arrangement of insulation in the wall. For example, consideration may be given to the two walls shown in Fig. 1 in which the same insulation has been installed in two different positions. In the left arrangement the outer 1⅝ in. of the studding is exposed to air at a low temperature while in

* Assistant Research Officer, Division of Building Research, National Research Council of Canada. Junior Member of A.S.H.V.E.

** Professor of Mechanical Engineering, University of Saskatchewan. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

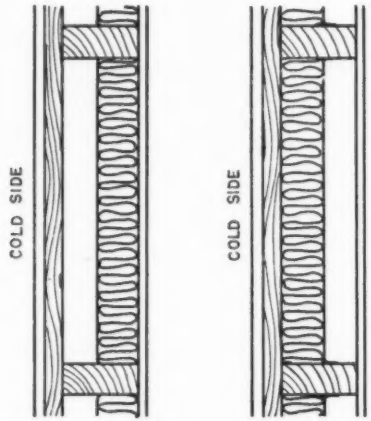


FIG. 1. TWO METHODS OF INSTALLING BATT INSULATION

the right arrangement the entire stud is in a region of much higher temperature. It would be expected, therefore, that a greater heat flow would occur laterally from the stud shown at left than that at right resulting in a lowering of surface temperature in the area over the stud.

When the vertical variations in temperature resulting from convection are also considered, it is seen that the heat flow is three dimensional and that the corresponding surface temperature pattern is complex, particularly in walls with blocking at mid-height. Since these variations are pertinent to any assessment of wall performance, the present study was extended to provide data on

TABLE 1—DESCRIPTION OF CURVES

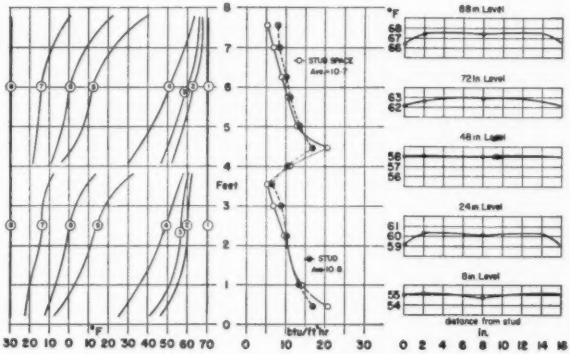
WALL PANEL NUMBER	DESCRIPTION OF INSULATION
1	Aluminum foil cemented to cold surface of plasterboard.
2	Single foil curtain creating two air spaces each faced one side with aluminum foil. Curtain sealed to surrounding framing members.
3	2-in. paper-enclosed mineral wool blanket insulation (actual thickness 1½-in.) creating two air spaces. Blanket sealed to surrounding framing members.
4	2-in. thick mineral wool batt insulation placed next to plasterboard to form one air space on cold side of insulation.
5	Same as 4 except batt placed against sheathing. Paper backing sealed to surrounding framing members.
6	Same as 5 except ⅜ × 14½-in. strips cut from paper backing at bottom and top of each air space.
7	Same as 2 with ⅜ × 14½-in. gap at top and bottom of each air space.
8	Same as 3 except blanket stapled but not sealed to surrounding framing members, staples 6-in. apart.
9	Same as 3 with ⅜ × 14½-in. gap at top and bottom of each air space.

the heat flow and surface temperature pattern at points other than at the center of the stud space.

This project was conducted as part of a program of Cold Weather Wall Research being carried on cooperatively by the University of Saskatchewan and the Division of Building Research, National Research Council, at Saskatoon.

TEST APPARATUS

The basic test apparatus employed has been described previously.[1] Closer regulation of air temperature on both sides of the panel was obtained by the use of resistance-thermometer bridge control systems in place of those originally used. This change involved no alteration in the air heating system on the



Single foil curtain creating two air spaces—
Foil sealed to surrounding framing members

FIG. 2. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 2

warm side of the panel, but an intermittent electrical reheat system was substituted for the bypass damper arrangement formerly employed in the cold room.

A second track and carriage was installed on the baffle facing the warm side of the test panel opposite a stud location to permit simultaneous heat meter traverses vertically over a stud as well as between studding. Additional thermocouples were installed to measure wall surface temperatures over the stud and 2 in. from the stud at five different heights, at locations not immediately adjacent to the metal screws used in construction of the panel.

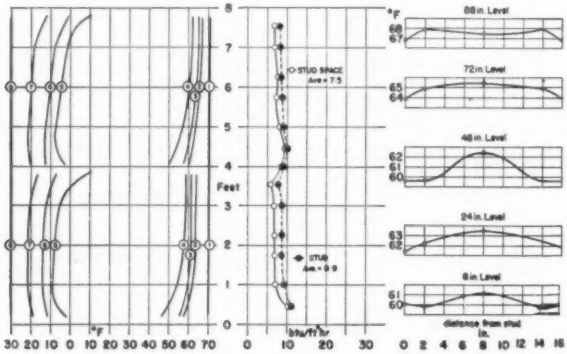
The heatmeters employed were of the multiple differential thermocouple type, with the sensitive elements mounted between synthetic resin sheets to form a finished meter $4\frac{1}{2} \times 4\frac{1}{2} \times 3/64$ in. thick. The meter used to traverse the wall at the stud contained two thermopiles, each measuring $\frac{3}{8} \times 1\frac{1}{2}$-in. spaced $1\frac{3}{8}$-in. on centers. The meter was oriented with the $1\frac{1}{2}$-in. dimension of the elements horizontal so that no portion of the thermopiles extended beyond the stud. The meter used to traverse the area at the center of the stud space con-

tained four thermopiles each measuring $\frac{3}{8} \times 2$-in. equally spaced, within an area $2 \times 2\frac{5}{8}$-in.

TEST PROCEDURE

The same basic wall panel, 8 ft in height by five standard stud spaces in width, was used for all tests. As in the previous study, only the center three stud spaces were actually employed as test area, the outer two spaces being fully insulated with mineral wool insulation.

The test procedure was essentially the same as for the tests reported previously, except that cold room air temperatures were maintained at a more



Two-inch mineral wool blanket insulation, creating two air spaces, sealed to surrounding framing members

FIG. 3. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 3

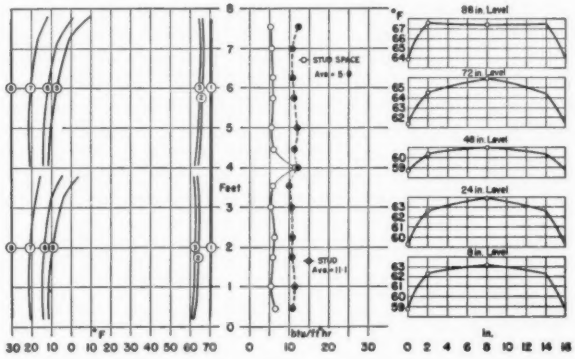
nearly uniform value, and heatmeter indications were recorded for a slightly longer period of time at each location.

DESCRIPTION OF WALLS TESTED

The basic wall panel consisted of $\frac{3}{8}$-in. plasterboard inside, on 2×4-in. ($1\frac{1}{2} \times 3\frac{5}{8}$-in. actual) studding spaced 16-in. on centers, $25/32 \times 10$-in. spruce shiplap sheathing, building paper and 6- by $\frac{1}{2}$-in. bevel cedar siding on outside. The exterior siding and sheathing, and interior plasterboard were secured with wood screws and the siding was given three coats of oil paint. Joints between framing members and sheathing were sealed with masking tape.

A total of nine different wall constructions was studied, each with blocking at mid-height. Only three different types of insulation were used, but variations were introduced in certain details of application. The various panels tested are described in Table 1.

Walls Nos. 1, 2, 4, 5, 6 and 7 were similar to walls studied in the previous paper, except for the insertion of horizontal blocking at mid-height. The paper-



Insulated with two-inch mineral wool batt—Air space on cold side of insulation

FIG. 4. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 4

enclosed blanket insulation installed in Walls 3, 8 and 9 was not used in the previous study.

TEST RESULTS

Test results are presented graphically in Figs. 2 to 10. The vertical variations in temperature at various locations through the wall at the center line of the stud space have been plotted at the left hand side with the numbering system employed for identification of the curves as shown in Table 2.

The vertical variations in heat flow rates into the wall at the centerline of the center stud space and at the stud immediately to the left have also been plotted for each wall. The values shown were the average of at least two runs. The heat flow rates given were obtained from observed heatmeter millivoltages using the conversion data provided by the manufacturer, without further calibration. When the traverse of Wall No. 1 was repeated with the two meters

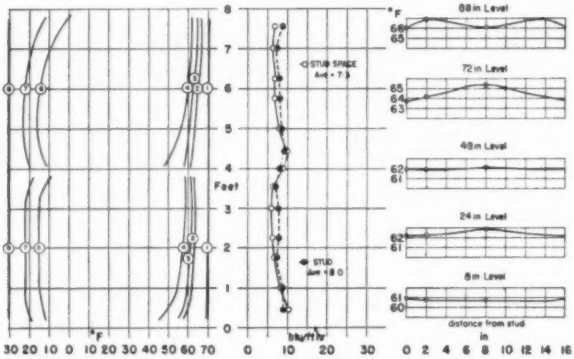
TABLE 2—IDENTIFICATION OF CURVES ON FIGS. 2-10

CURVED NUMBER	LOCATION
1	Warm side air 1 in. from wall surface.
2	Warm side of plasterboard.
3	Cold side of plasterboard.
4	Air space 3/8 in. from plasterboard.
5	Air space 3/8 in. from sheathing.
6	Warm side of sheathing.
7	Cold side of sheathing.
8	Cold side air 2 in. from wall surface.

interchanged the average percentage difference between corresponding readings was found to be within 2 percent.

Temperatures of the wall surface at locations over the stud immediately to the right of the center of the wall, 2 in. and 8 in. to the right of this stud are given in Figs. 2 to 10 for five different elevations. These measurements were, therefore, associated with the stud space adjacent to that in which the heatmeter data and additional temperature measurements were made.

Precautions were taken to prevent entry of water vapor into the warm room even though ambient conditions outside in the laboratory were of the order of



Insulated with two-inch mineral wool batt—Air space on warm side of insulation—Batt sealed to surrounding framing members

FIG. 5. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 5

15 percent relative humidity or lower for most of the test period. Further protection against condensation was provided by painting the cold side of the plasterboard with two coats of aluminum paint followed with two coats of white oil paint. In spite of these precautions, some frosting occurred on the sheathing in all tests.

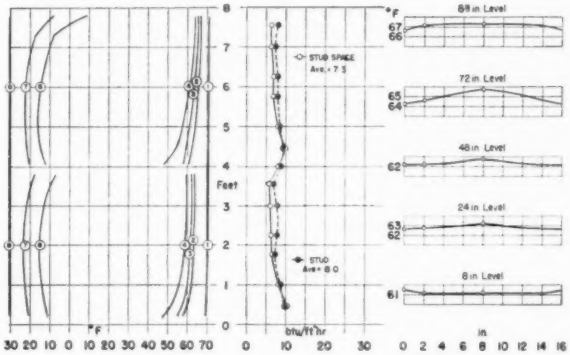
HEAT FLOW INTO WALLS WITH AIR SPACES

The effect produced by inserting horizontal blocking at mid-height was such as to create similar patterns of temperature and heat flow in the sections of the wall above and below the blocking. Each of these patterns was remarkably like those found over the full height of corresponding walls previously tested without blocking. All the walls with air spaces immediately behind the plasterboard showed rates of heat flow into the wall decreasing with height for each air space. This variation was in keeping with the temperature conditions throughout the wall, temperatures generally increasing with height for each space.

Of the walls in which insulation was ideally applied, those containing reflective insulation exhibited the greatest vertical variation in rate of heat flow into

the wall. This feature, more fully discussed in the previous paper, can be attributed to the predominance of the effects of convective heat transfer with the reduction in radiant transfer across the space.

The openings left in the reflective curtain at the top and bottom of each air space in Wall No. 7 increased the average heat flow rate into the area between studding by 37 percent over that for Wall No. 2, but the variation with height was not as extreme. This may partially be accounted for by the fact that condensation had formed on the reflective curtain in Wall No. 2 during test, on an area extending approximately 8 in. from the bottom of each air space. This



Insulated with two-inch mineral wool batt—Air space on warm side of insulation—Gaps of ⅜ in. cut in paper backing at top and bottom of each air space

FIG. 6. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 6

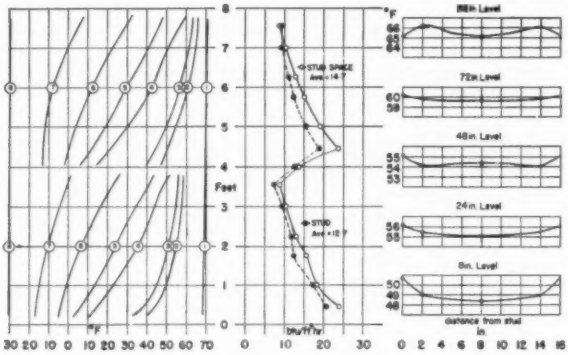
condensation undoubtedly affected the emissivity of the foil in this region and increased the heat flow into the wall.

The effects of air-space convection are apparent from a comparison of heat flow rates into the center of the stud space for Walls Nos. 3, 8 and 9 in which a mineral wool blanket insulation was installed. Wall No. 9, in which openings existed through the insulation at the bottom and top of each air space, showed the greatest vertical variation in heat flow of all the walls tested, the rate of heat flow into the wall at the 5-in. level being 3 times the average. In addition the average rate of heat flow increased by 21 percent of that for Wall No. 3 in which the insulation was sealed in place. When the insulation was stapled, but not sealed to the surrounding framing members, as in Wall No. 8, only a slight increase in average heat flow over that for Wall No. 3 was observed, with the vertical variation being somewhat greater.

No significant differences in thermal characteristics were found between Walls Nos. 5 and 6 although the insulation was sealed in place in Wall No. 5 and ⅜- by 14½-in. openings were cut through the paper backing of the insulation at the top and bottom of each air space in Wall No. 6. This close agreement suggests that little or no air movement occurred through the insulation.

The results shown here, as in the previous paper, indicate that the location of semi-thick batt-type insulation affects both the variation in heat flow with height and the average heat flow rate at the center of the stud space. The greater variation in heat flow vertically in Wall No. 5 can be considered as due to convection in the air space behind the plasterboard.

The presence of this air space makes it possible for heat to enter at one point and leave at some higher point. It is equally possible, however, for heat to enter the convective air stream through the area between studs and be transported laterally to, and leave by, the stud path. This can account for the higher average heat flow between studs for Wall No. 5 and is, in fact, supported by the lower readings of heat flow into the wall opposite studs. When no air space is present on the warm side of the insulation, only relatively high resistance paths to heat flow in the plane of the wall are provided until the air space on the cold



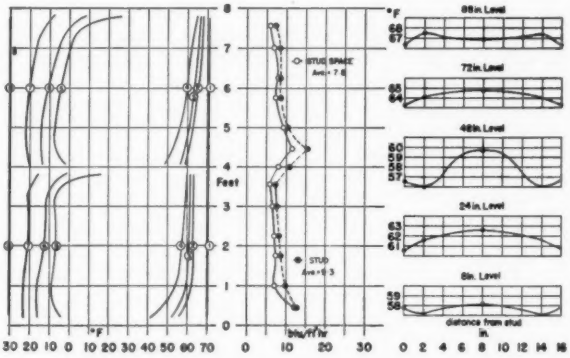
Insulated with single foil curtain creating two air spaces —Gap of ⅜ in. at top and bottom of each air space

FIG. 7. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 7

side of the insulation is reached. It follows that some of the heat enters the outer air space in Wall No. 4 by way of the stud path and leaves through the sheathing between studs.

In all the walls studied, the vertical variation in the rate of heat flow into the area over studding was very similar to the variation existing in the area between studs. Since such vertical variations can only be attributed to the change in air space temperature with height, it is obvious that the rate of heat flow into studding is influenced to a great extent by the air temperature conditions to which the sides of the stud are exposed. This provides further evidence of the importance of lateral heat exchange between studs and air spaces in determining the overall pattern.

Further consideration might be given to comparison of measured heat flow rates into studding and into the stud space for the walls tested. Some doubt exists, however, as to the calibration corrections to be applied to heatmeter readings when there has been a change in the arrangement of material on which



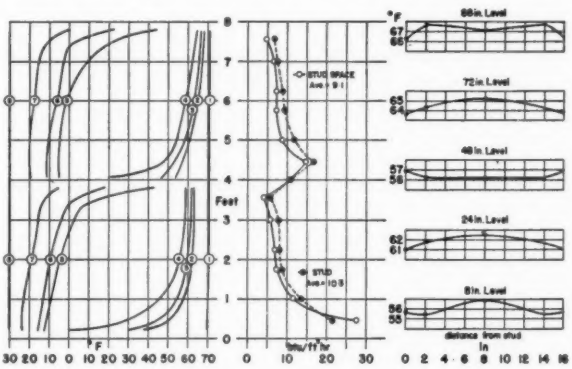
Insulated with two-inch blanket creating two air spaces—
Blanket stapled but not sealed to surrounding framing members

FIG. 8. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 8

the heatmeter is placed. Such comparison on a strict quantitative basis has, therefore, been reserved until these features of the heatmeter technique have been evaluated, but there seems to be no reason to question the overall pattern of heat flow which the uncorrected values imply.

SURFACE TEMPERATURE DISTRIBUTION

Inside surface temperatures of exterior walls have a direct bearing on some aspects of indoor comfort conditions, on the performance of panel heating



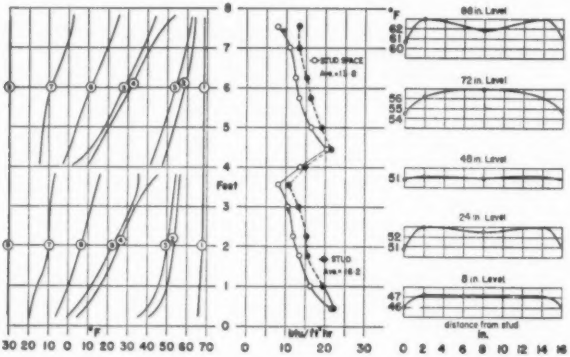
Insulated with two-inch blanket creating two air spaces—
Gaps of 3/8 in. at top and bottom of each air space

FIG. 9. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 9

systems, on surface condensation problems and on the formation of dirt patterns on walls. It is of some importance, therefore, to note the very substantial deviations of actual surface temperature patterns from those implied by simple unidirectional heat flow theory.

Nielsen[2] has discussed the problem of dust marking and has shown it to be due to differences in rate of accumulation on adjacent areas at different temperatures. Rogers[3] suggests that a difference of from 3 to 5 F deg in wall surface temperatures may produce noticeable dust patterns. It may be noted that using this criterion, Wall No. 4 may exhibit dust marking over studding, while all the walls except Nos. 4, 5 and 6 may exhibit dust marking just above horizontal blocking. It is not correct, therefore, to assume that the application of insulation will necessarily eliminate the tendency to noticeable dust marking.

Surface condensation will occur whenever the wall surface temperature falls below the dew point temperature of the air. Potential areas for surface con-



Aluminum foil cemented to cold surface of plaster board

FIG. 10. TEMPERATURE AND HEAT FLOW VARIATION IN WALL NO. 1

densation will invariably be found at the lower portion of a wall. When the wall construction is such as to produce a high variation in surface temperature vertically over the area between studs the coldest area will tend to be at the bottom of this vertical zone. When, however, the wall arrangement is such as to produce high heat flow into the studs, the convection effects over the center of the space will not be so marked and the coldest area will be opposite the studding. The effect of leakage between air spaces is to exaggerate the vertical variation in surface temperature between studs. All the walls except No. 4 show predominating convective effects and in none of these is the area over studs markedly colder than the area between studs. Wall No. 4, however, has small vertical variations in surface temperature, since the convective effects are restricted to the cold side of the insulation. For the same reason, the heat flow through the studding tends to be high and the coldest area is to be found opposite studding.

Wall No. 4, showing the greatest horizontal variation in surface temperature at the lower portion of the wall does not represent the worst possible

case of such variation. In many practical installations, the insulation will not be installed to fit tightly against the studs for its full thickness and relatively lower surface temperatures over studs may be expected as a result of the increased heat flow through studs. The most extreme case known to the authors is for a form of insulation not represented here where leakage existed and the convective effects combined with high heat flow through the studs to produce, not only high variations in temperature vertically, but also extremely low temperatures over studs.

It may be noted that although Wall No. 4 exhibited the greatest horizontal variation in surface temperatures over the lower portion of the wall, Walls Nos. 1, 2, 7 and 9 exhibited much lower surface temperatures in this region. The lowest surface temperature measured at the 3-in. level on Wall No. 9 was 39 F which would permit an inside relative humidity of only 32 percent at 70 F before condensation occurred. This applies to the test conditions of an inside to outside air temperature difference of 100 F deg which although severe for most of the United States, is not unrealistic for the northern part of the continent. All tests, however, were carried out with room air temperatures maintained reasonably constant over the height of the wall. In actual practice some further lowering of surface temperatures at the lower portions of walls may result from reduced room air temperatures at floor level or from contact with cold floors or foundations.

CONCLUSIONS

The following conclusions have been drawn from the test results and observations:

1. The heat flow and surface temperature patterns in frame walls are greatly influenced by movement of heat in the plane of the wall.
2. Substantial movement of heat in the plane of a wall occurs largely as a result of convection in air spaces coupled with heat interchange between air spaces and framing members.
3. Vertical variations in both temperature and heat flow are related to the convection in air spaces and can be exaggerated by air leakage into or between air spaces.
4. Horizontal variations in both temperature and heat flow are related not only to the type but also to the arrangement of the material used for insulation, and particularly to the thermal resistance provided over the area between studs before an air space is reached.
5. The characteristic pattern of temperature and heat flow resulting from convective effects is not greatly changed by reducing the free height of the air space from 8 ft to 4 ft and in a wall with horizontal blocking at mid height will be substantially repeated over each blocked-off area.
6. Blocking at mid height may produce potentially serious conditions for dust marking in the area immediately above blocking.
7. The improvement to be obtained through the use of insulation, in tendency toward dust marking and surface condensation depends not only on the insulation but also on the thermal properties of the overall arrangement created.

ACKNOWLEDGMENT

The authors wish to acknowledge with thanks the assistance given them in the taking and recording of data by other members of the staff of the Prairie Regional Station, Division of Building Research, *National Research Council.*

REFERENCES

1. Thermal Performance of Frame Walls, by G. O. Handegord and N. B. Hutcheon (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 171).
2. Dirt Patterns on Walls, by R. A. Nielsen (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 247).
3. Design of Insulated Buildings for Various Climates, by T. S. Rogers (F. W. Dodge Corp., 1951).

DISCUSSION

R. K. THULMAN, Washington, D. C.: The authors are to be congratulated on the competence with which they have carried out this investigation.

The use of insulation in the walls of residences has lagged far behind the use of insulation in ceilings. The actual amount of insulation used in walls is also less than that used in ceilings. It is important, therefore, that the application of wall insulation be guided by sound technical data.

The data in this paper show that the variation in the coefficient of heat transmission in relation to the height of the wall is greater as the effectiveness of the insulation decreases. The tests were conducted with a uniform temperature of the air on the warm side of the wall. Actually, a uniform temperature is rarely found, there being a decided gradient increasing from floor to ceiling. Common types of heating systems often produce gradients of 10 to 20 deg, sometimes as high as 40 deg.

The fact that this work was done with Canadian climatic conditions in mind should not blind us to the possibilities of the direct application of the results to construction in this country. Here Canadian conditions are approached by the large gradients produced by the less effective heating systems used in the United States.

D. L. MILLS, Rome, N. Y.: When I read papers of this kind I wish that the information might be translated into layman's language and published for the benefit of builders and building contractors. In Fig. 1 valuable practical information comparing the results of two methods of installing insulation is presented, but unfortunately many builders and contractors are not acquainted with these facts.

Many contractors and builders with whom I am acquainted have difficulty interpreting graphs. The translation of the information contained in this paper into handbook language would be a valuable contribution to the industry.

G. L. TUVE, Cleveland, Ohio: I am impressed with the difference in heat transfer and temperature gradient that the authors reported in connection with incidental or accidental spaces left in the insulation construction, and I would like to ask whether you have real success in getting contractors to install these insulations as desired in Canada. In Cleveland during the last 10 years I have built two houses and only by being on the job and watching what was going on was I able to obtain satisfactory insulation installation.

It seems to me great differences are occurring in residential construction, in particular due to method and care in installing the insulation. Is there not some way in which members of the Society could exert influence to improve on the actual practices in the field?

W. A. DANIELSON, Raleigh, Tenn.: My experience in the study of temperature gradients within the room indicates that they are proportional to the heat loss of the building, other considerations being equal, *i.e.*, if you have a room with a high heat loss, you will have a high temperature at the ceiling and a low temperature at the floor level with resultant drafts. I wonder if insulation could be applied on the side

of the stud, along the wall, and along the next stud. In that way, the stud would be insulated from the air space.

W. O. HUEBNER, New York, N. Y.: I was very pleased to have Mr. Handegord mention the problem of smudging in his discussion of the surface temperature distribution. Everybody concerned with air distribution is also concerned with this problem, which is of great importance but is too often neglected.

D. B. ANDERSON, St. Paul, Minn.: Professor Tuve mentioned the problem that we have in educating contractors in the proper application of insulation. I believe that many manufacturers are cognizant of this situation. A number of these manufacturers are doing an extensive job in educating the contractors through the use of specially prepared films, literature, contractor meetings, etc., in which the proper application of insulation is stressed.

AUTHORS' CLOSURE (G. O. Handegord): The authors appreciate the interest shown by the discussors in presenting their comments on this paper.

We are concerned with the variations in heat flow and temperature which have been measured in these tests since they represent significant departures from normally assumed uniform conditions. These variations become more serious with lower outside temperatures and, under the severe winter conditions of Western Canada, may create numerous problems.

Mr. Thulman has drawn attention to the fact that air temperatures were maintained nearly uniform over the warm side of the wall during all tests. Forced circulation of air over the panel was necessary to achieve this uniformity and in this respect the test conditions are not representative of actual practice.

We would certainly agree with Mr. Thulman that such uniform air temperatures are seldom experienced in actual buildings. The usual air temperature gradient from floor to ceiling found in actual houses will generally tend to produce still lower surface temperatures near baseboard level. In some cases the combined effects of room air gradient and convection in the air spaces in the wall will be such as to produce serious condensation problems.

One particular case in our experience involved a basementless house, heated by an oil fired space heater, having an uninsulated floor over a crawl space and inadequately insulated walls. These factors combined to produce a severe floor-to-ceiling air temperature difference, temperature ranging from 55 F at floor level to 105 F at the ceiling when the outside air temperature was only 0 F. The wall insulation was of a type which allowed marked vertical variations in temperature to exist through air space convection and also permitted high heat flow through studding. These features, combined with the room air temperature variation, resulted in frost formations on the lower part of the wall which extended in peaks three feet up the wall at each stud.

Mr. Mills has suggested that this information be presented in a manner more readily understood by contractors and insulation applicators. The Division of Building Research will be publishing a Better Building Bulletin on insulation in the near future, which will incorporate some of the general conclusions of this study. He mentioned in particular the alternative methods of placing 2 in. thick batt insulation in frame walls. The results presented here suggest that more uniform horizontal surface temperatures will be experienced when the batt is placed against the outside sheathing. The overall thermal resistance of a wall insulated in this manner, however, appears to be slightly less than a wall in which the batt is placed immediately beneath the inside finish. This feature is perhaps one of quite a number that must yet be considered in comparing the two methods of placement.

The difficulty of insuring correct application of insulation has been pointed out by Professor Tuve. We have endeavored, whenever possible, to impress contractors in

our region with the importance of correct installation, but have not been entirely successful. Careful application is probably most important in the case of blanket or foil type insulation installed to create multiple air spaces. These types of insulation are not too common in the prairie region of Canada, and in many cases bulk insulations or full-thick batts are used, in which case the problems arising through incorrect applications are not so serious.

Mr. Huebner has mentioned the problem of dust marking due to surface temperature variation. We are very conscious of this problem in Western Canada since our severe winter climate promotes more extreme variations in surface temperature and hence more noticeable dust patterns. I would like to emphasize that some of the test results which have been obtained suggest that in certain cases the application of insulation may not reduce the tendency toward dust marking but may, in fact, increase such tendency, particularly if the insulation is not carefully installed.

1488

No. 1488

DEVELOPMENT OF THE THERMAL CONDUCTIVITY PROBE

By F. C. HOOPER* AND S. C. CHANG**, TORONTO, ONT., CANADA

A TRANSIENT heat flow apparatus for the rapid determination of the thermal conductivity of solids has been under development at the University of Toronto for several years. In an initial paper[1] the apparatus was described, and the technique of its use and the history of its development were outlined. Since that time the probe has found application in several laboratories and in many field investigations. Research and development work has continued at Toronto.

The present paper will make available some of this experience in application of the probe and present a newly developed chart for the graphical evaluation of thermal conductivity from the probe readings.

APPLICATION EXPERIENCE

The thermal conductivity probe is basically a linear heat source of constant strength which can be inserted in the material to be tested. The time and temperature rise relationship can be determined and used for the evaluation of the thermal conductivity of the surrounding material.

Many forms and sizes of the instrument have been used, but the construction and dimensions described in the original paper, as shown in Fig. 1, have proved to be generally satisfactory where test specimens of suitable size are available. This form of the instrument is now regarded as standard.

Experience has revealed the importance of several practical considerations. Mechanically, a weak point exists at the point of flexure where the leads leave the probe, and reinforcement of this position by elastic binding tape or other means is necessary if short-circuits and leakages are to be avoided. Care must be taken to avoid the use of lead-in wire of heavier gage than necessitated by

* Lecturer in Mechanical Engineering, University of Toronto. Junior Member of A.S.H.V.E.

** Research Assistant, Department of Mechanical Engineering, University of Toronto.

[1] Transient Heat Flow Apparatus for the Determination of Thermal Conductivities, by F. C. Hooper and F. R. Lepper (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 309).

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

the electrical requirements, since a considerable conduction error due to end loss can arise in specimens of low conductivity. Care should also be exercised in installing the heater wire to avoid local straining with its consequent non-uniform heating. When storage batteries are used as a source of power as in portable forms of the instrument, it is important to provide sufficient capacity to avoid progressive voltage drop and the consequent progressive fall in heat source strength over the period of the test.

The standard probe has been used in materials ranging from dense moist clay to rock-wool fiber insulating material of very low density. No lower

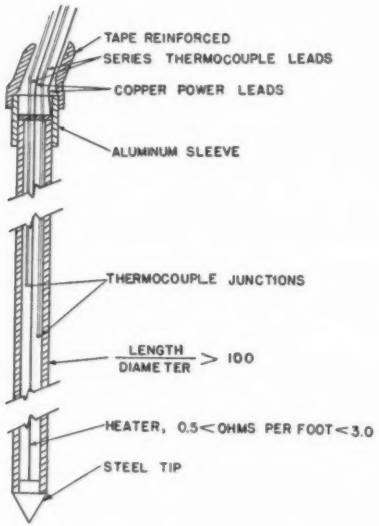


FIG. 1. THE STANDARD FORM OF THERMAL CONDUCTIVITY PROBE

limit to measurable thermal conductivity has become apparent, but an upper limit is imposed by the sensitivity and speed of the temperature measurement system employed, since temperature rises become smaller and significant time intervals become relatively shorter as conductivities rise. Moreover, deviations from ideal behavior are more significant at short time periods.

Several special forms of the probe have been developed for specific uses. A small probe, 4 in. in length and 0.060 in. in diameter, suitable for use in small specimens, was built and applied in measurements on dairy products. The minimum size of probe is apparently restricted only by the craftsmanship required in the construction of probes of adequate length to diameter ratio with small overall lengths.

In undisturbed ground the moisture content, and consequently, the conductivity, vary considerably with depth. A special form of the probe has been

used in the advance exploration of the projected routes of heated oil pipe lines and buried electric cables, and in the testing of heat pump ground coil sites. These probes are 5 to 8 ft in length and of stainless steel tubing approximately $\frac{3}{8}$ in. in diameter. Thermocouples are placed at intervals of about 1 ft over the length and are read separately and simultaneously. Conductivities are calculated separately for each thermocouple position. The probe is driven vertically along a path prepared by driving and pulling a hardened steel rod of the probe diameter. Fig. 2 shows typical values taken at a heat pump site.

At least two special factors enter into this application, both due to the vertical variation of properties in the soil. Since there is a temperature gradient

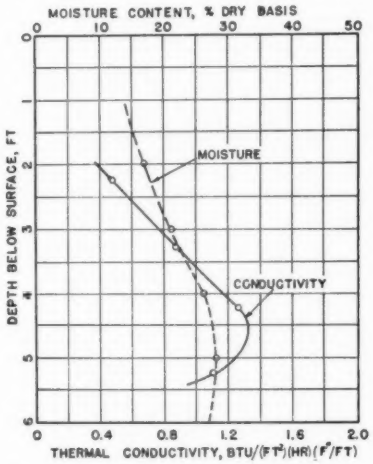


FIG. 2. MOISTURE AND CONDUCTIVITY VARIATION WITH DEPTH IN AN UNDISTURBED SANDY SOIL

with depth over the length of the probe, some longitudinal conduction must take place in the probe wall. Moreover, the heat flow from the probe is not entirely in a radial plane because both the soil temperature and conductivity vary with depth. Both of these effects must introduce errors. Fortunately high precision is not normally required in this type of measurement, and the values obtained from it are apparently satisfactory for the purposes mentioned.

In the laboratory, considerable work has been done using a single straight wire, not covered by a tube, as the heat source, with thermocouples mounted on or near the wire. This simple device, while only suitable for laboratory work, gives satisfactory results in moderately homogeneous material, and is particularly sensitive in very low density specimens.

Typical results from tests on various materials with several forms of the probe are shown in Table 1. It will be noted that the range of variation of the thermal conductivities measured was more than fifty-fold. In this table

the significant time shown indicates period of observation used in the computation of the result.

METHOD OF ANALYSIS

The temperature rise of an infinite line heat source of constant strength and zero diameter in an infinite, initially isothermal, homogeneous specimen is given by Equation 1:[1]

$$\theta = \frac{Q'}{2\pi k'} \int_{rn}^{\infty} \frac{e^{-\beta^2}}{\beta} d\beta = \frac{Q'}{2\pi k'} [I(rn)] \quad \ldots \ldots \ldots \quad (1)$$

where

k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit per inch).
k' = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).
n = a function $\frac{1}{2}(\alpha t)^{1/2}$.
Q = heat input per unit length, watts per foot.
Q' = heat input per unit length, Btu per (hour) (foot).
r = radial distance of the position of temperature measurement, feet.
t = time elapsed from start of heating, hours.
$I(rn)$ = a function,

$$\int_{rn}^{\infty} \frac{e^{-\beta^2} d\beta}{\beta}$$

α = thermal diffusivity, square feet per hour.
β = an integral parameter.
θ = temperature rise after start of heating, Fahrenheit degrees.

It is apparent that an actual probe cannot behave in exactly the manner of this ideal line heat source. The probe is finite in diameter and length, and is not homogeneous throughout its cross-section. It is made of a different material than the test specimen. Moreover, temperature is measured by a finite thermocouple, and the precise position at which it measures temperature is not determinable.

While it is possible to compensate by direct mathematical methods for some of the foregoing deviations, such as for finite probe diameter, other factors such as non-homogeneity of the probe, are not susceptible to direct mathematical treatment. Adjusting mathematically for only some of the deviations does not necessarily yield a better approximation, as these may be compensated, or even over-compensated, by the remaining factors. Failure to appreciate this can lead to invalid conclusions.

There are at least two effective methods which permit allowance for the non-ideal behavior of the actual probe without abandoning the simple line heat source assumption. Originally a time scale adjustment method was used. This method is given in detail in the original paper.[1] Further examination of the problem has since indicated that adjustment of the assumed position of temperature measurement will yield results which are at least as reliable. In practice the latter has proved to be a much more convenient method of adjustment.

Both of these methods have the effect of adjusting the lower limit of the integral in Equation 1. Time adjustment alters n which is a function of time. Position adjustment alters r, the assumed radius of the temperature measurement.

The method of introduction of the radius adjustment becomes apparent when the relationships which follow are considered.

The value of θ from Equation 1 can be obtained for any corresponding value of (rn). A series of such values may be chosen and a family of linear lines of slope $Q/(2\pi k)$ is obtained when plotted as in Fig. 3 (B).

Variation of I (rn) with (rn) can be shown graphically as in in Fig. 3 (C).

Further, the variation of (rn) with time can be shown for all values of the parameter $r/\alpha^{\frac{1}{2}}$, as in Fig. 3 (D).

The observations of temperature and time made during an actual probe test can be plotted as in Fig. 3 (A).

If the value of Q/k is known, then for any point A_1 on the test curve there is corresponding temperature rise from which may be obtained the corresponding values of $I(rn)$, and consequently of (rn) and of $r/\alpha^{\frac{1}{2}}$, shown in the sequence of operations A_1, B_1, C_1, D_1 in Fig. 3. Repeating this sequence for points A_2, A_3, A_4, etc. will give a corresponding series of points D_2, D_3, D_4, etc. on Fig. 3 (D).

If a suitable value of Q/k has been chosen, and if the assumption of radius adjustment is justified, then a curve of constant $r/\alpha^{\frac{1}{2}}$ must be obtained. This is indicated by a curve conforming to the contours of the Fig. 3 (D) lines.

TABLE 1—TYPICAL RESULTS FROM PROBE DETERMINATIONS

MATERIAL	DENSITY lb/cu ft	TYPE OF PROBE	BASE TEMP OF SPECIMEN F DEG	OVERALL TEMP RISE F DEG	SIGNIFICANT TIME MIN	CONDUCTIVITY[a]
Silica aerogel..	4.55	12 in. long, 3/32 in. dia, brass tube	73	22.5	4.5	0.125
Rockwool.....	10.2	12 in. long, 3/32 in. dia, aluminum tube	78.1	25.1	7	0.263
	10.2	17.5 in. long single 24 Ga wire with thermocouple attached	72.4	18.3	6	0.267
Corkboard....	10.0	Single 24 Ga wire laid between two boards	71.8	22.3	5	0.290
Saw dust.....	11.0	12 in. long, 3/32 in. dia, brass tube	63.3	20.8	7	0.290
Wheat.....	33.0	7.5 in. long, 3/32 in. dia, aluminum tube	73.2	14.4	5	0.926
Butter.....	62.3	7.5 in. long, 3/32 in. dia, aluminum tube	38.4	8.8	5	1.37
Margarine....	62.4	Single 24 Ga wire, with thermocouple attached	39.9	8.1	5	1.62
		Single 24 Ga wire, with thermocouple offset 1/8 in.	45.6	2.3	5	1.61
Sand.....	100.7 1.38% moisture	12 in. long, 3/32 in. dia probe, with 3 thermocouples in series	71.5	40.7	8	3.01
Humus Soil...	15.7	12 in. long, 3/32 in. dia probe, with 3 thermocouples in series	76.6	27.5	7	3.62
Concrete 1:2:2 mix.....	2 days old	12 in. long probe, 3/32 in. dia, cast into specimen	82.8	26	5	10.6
	19 days old		71.4	35.4	6	7.71
	31 days old		68.4	42	6	7.06

[a] Btu per (square foot) (hour) (Fahrenheit degree per inch).

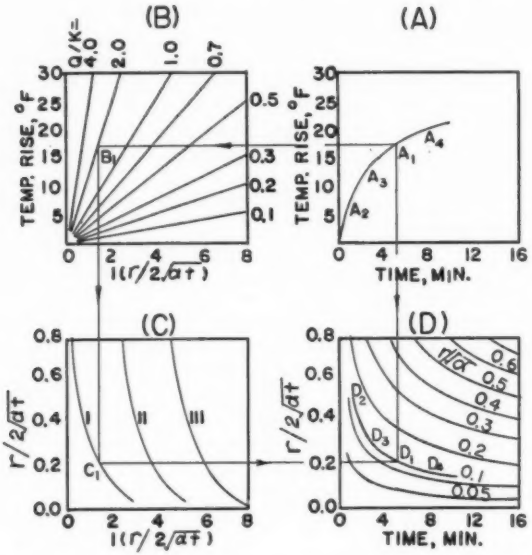


FIG. 3. GRAPHICAL PRESENTATION OF THE MATHEMATICAL LINE HEAT SOURCE RELATIONSHIP

For convenience the four charts of Fig. 3 can be superimposed to form a single chart as in Fig. 4.

In using this method of solution it is necessary to select arbitrarily a value of Q/k and then proceed by successive approximation. For example in Fig. 5, for the test curve as seen in the figure, a value of $Q/k = 3$ was chosen, and yielded curve (a), which proved too shallow. Curve (b), for the second chosen value of $Q/k = 1$, proved too steep. A final choice of $Q/k = 2.1$ gave line (c), which conforms very closely to the contours. Since the value of Q is known from the experiment, the value of k has thus been determined as $Q/2.1$.

The precision of reading is reduced when the $r/\alpha^{\frac{1}{2}}$ line obtained lies close to the axes. To permit equal precision in the use of the chart (Fig. 4) over a wide range of values of Q/k, three lines I, II, and III, showing the $r/2(\alpha t)^{\frac{1}{2}}$

TABLE 2—THE NON-CONSTANCY OF THE EQUIVALENT RADIUS

MATERIAL	CONDUCTIVITY Btu/(hr) (sq ft) (F deg/ft)	DENSITY lb/cu ft	SPECIFIC HEAT Btu/(lb) (F DEG)	DIFFUSIVITY α, sq ft/hr	$\frac{r}{\sqrt{\alpha}}$	r ft
Silica aerogel.....	0.0104	4.55	0.20	0.0114	0.12	0.0128
Saw Dust.....	0.0243	11.0	0.40	0.00545	0.08	0.00592
Sand.....	0.223	95	0.20	0.0117	0.0066	0.000713

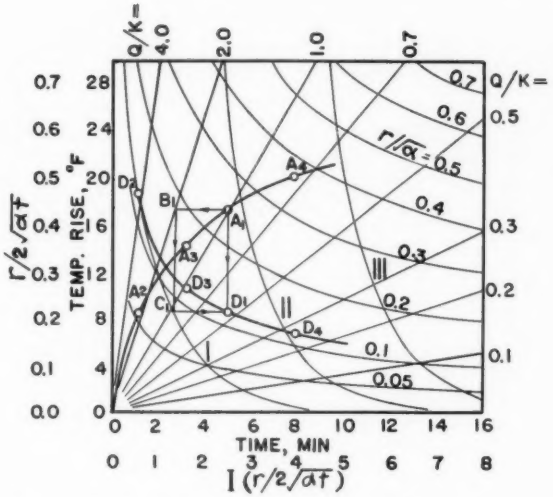


FIG. 4. COMPOUND CHART OBTAINED BY SUPERIMPOSING THE CURVES OF FIG. 3

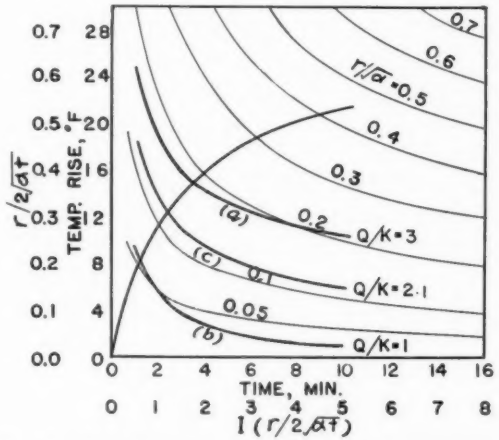


FIG. 5. THE SOLUTION OF A TYPICAL PROBLEM ON THE CHART

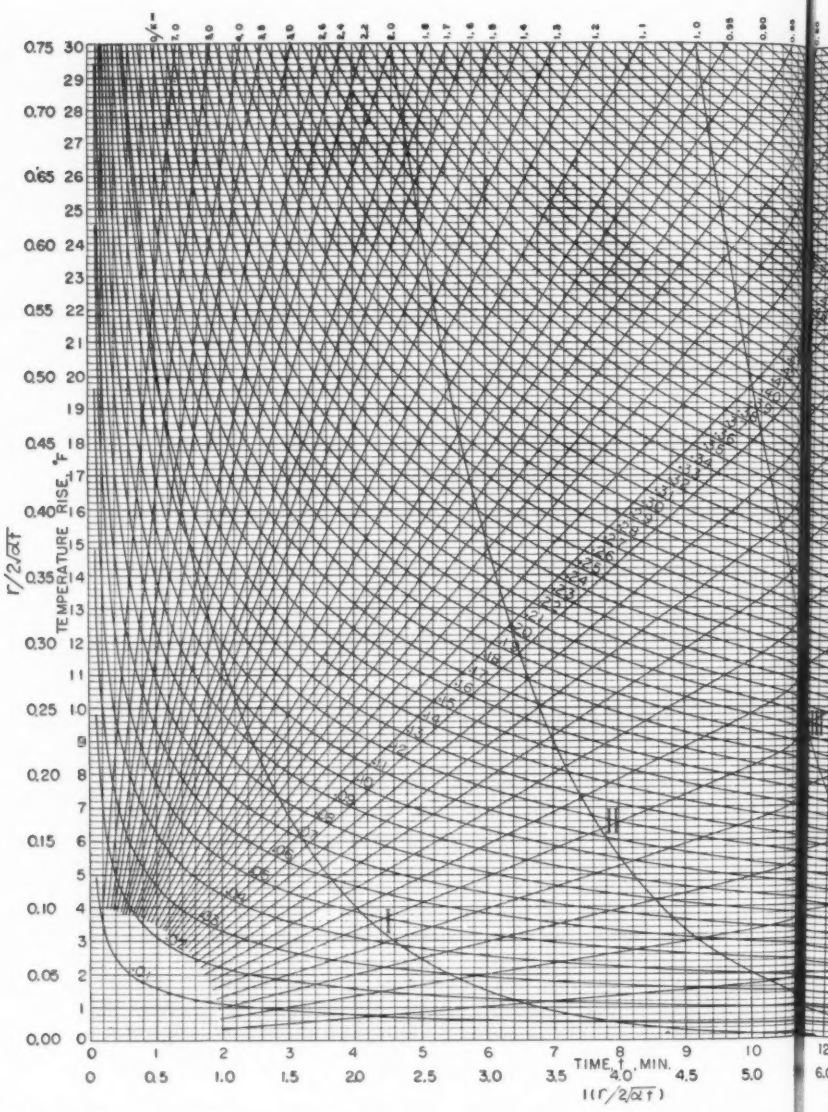
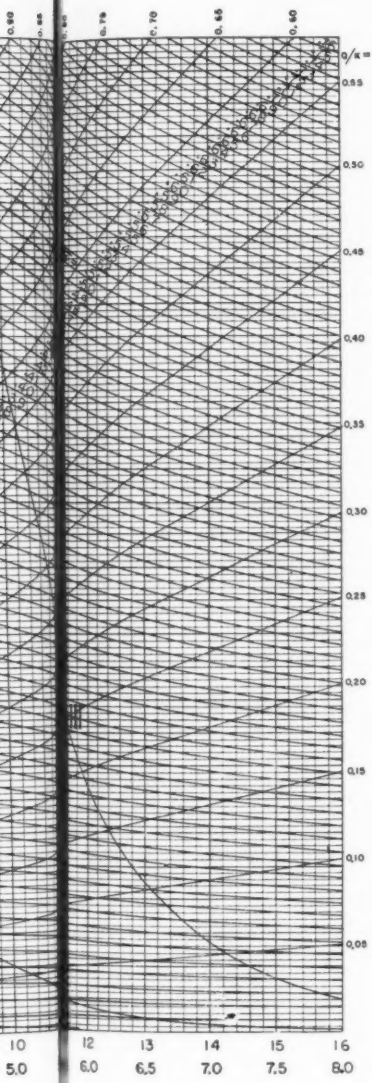


FIG. 6. A WORKING CHART TO FACILITATE THE EVALUATION OF THERMAL CONDUCTIV



NDUCTIVITY FROM PROBE OBSERVATIONS

function, have been provided. Actually, curve I corresponds to the scale values of $r/2(\alpha t)^{\frac{1}{2}}$ and $r/\alpha^{\frac{1}{2}}$ shown, while curves II and III correspond to scales expanded by factors of 10 and 100 respectively. In any problem the most appropriate of these lines is chosen. Since only the constancy of the $r/\alpha^{\frac{1}{2}}$ function is sought, the change from one of these curves to another does not alter in any way the procedure already described.

The 30 deg temperature rise range of the ordinate scale is convenient for most tests. Where temperature rises are too great or too small to be plotted conveniently to this scale, the temperature rise scale can be adjusted and the Q/k scale will be altered in the same proportion. That is, if the temperature rise scale is reduced by a factor of 2 to give a full scale value of 60, then the Q/k values will be altered by the same amount, so that $Q/k = 7$ would be read $Q/k = 14$.

For convenience in the practical calculation of probe results, the values of Q/k shown are expressed in terms of Q in watts per foot of probe length and k in Btu per (hour) (square foot) (Fahrenheit degree per inch).

Fig. 6 shows the chart, which is of the same form as Fig. 5, but of sufficient size and detail for use in actual calculations.

SIGNIFICANCE OF CHART VALUES

The chart method, by reducing the solution to depend upon the constancy of the $r/\alpha^{\frac{1}{2}}$ parameter, and not upon actual values of r and α, eliminates the difficulty found in assigning a value to r. The method, in effect, makes the determination at the most suitable value of r for a particular test condition.

The basic premise is, that if a constant value of $r/\alpha^{\frac{1}{2}}$ can be obtained by this graphical method, then it follows that the test curve can be correlated closely by Equation 1. That is, the value of Q/k is uniquely determined by this means.

Thus, the applicability of the method depends upon whether or not a constant $r/\alpha^{\frac{1}{2}}$

value can be obtained. A considerable number of determinations, such as those for the tests shown in Table 1, indicate that a very nearly constant $r/\alpha^{\frac{1}{2}}$ can be found. Moreover, it has been found that thermal conductivities of dry materials determined by this method agree well with values obtained on the hot plate apparatus.

If the method is not applicable under certain test conditions, or if observations have been incorrectly made, a constant $r/\alpha^{\frac{1}{2}}$ will not be obtainable. Thus, improper application is automatically avoided.

It should be noted that the value of r in the parameter $r/\alpha^{\frac{1}{2}}$ does not coincide with any fixed physical position in relation to the probe. This is an equivalent mathematical value only, and varies greatly with test conditions and materials. It is therefore not possible to use the $r/\alpha^{\frac{1}{2}}$ value found from the chart for the determination of α, because the value of r cannot be known. This non-constancy of r can be seen in the results shown in Table 2, which was constructed from test results from the same probe used in three quite different materials of known physical properties. The k values and $r/\alpha^{\frac{1}{2}}$ values shown were obtained by the chart method, and the r values calculated by substituting the known value of α. It will be seen that r is by no means constant for the same probe.

CONCLUSION

The practical utility of the instrument has now been demonstrated under a sufficient range of conditions to warrant the statement that the device is no longer of experimental interest only, but is a practical tool available for direct engineering application. The chart method of calculation described here greatly facilitates the computation of results and increases the reliability of the method.

ACKNOWLEDGMENT

Most of this work has been carried out in the Heat Transfer Laboratory at the University of Toronto. The financial assistance of the University School of Engineering Research and of the National Research Council at Ottawa is gratefully acknowledged.

The authors wish also to acknowledge the contributions to the research and development work made by Messrs. C. P. Lentz, J. G. Takata, G. R. Masson. and F. R. Lepper.

DISCUSSION

L. R. INGERSOLL*, Madison, Wis.: I feel that the authors (and also Lepper of the earlier paper) should be congratulated on developing a simple workable thermal conductivity probe. It is something that has been needed for a long time.

Naturally I am interested in their use of the line source integral, in which development and application I have been somewhat concerned. However, as might be expected, they have found that this theory must be supplemented by various practical considerations which come out in actual experiment. The way in which they have woven these two considerations—theoretical and practical—together is ingenious. I predict a large field of usefulness for this new tool.

---

* Professor of Physics, University of Wisconsin.

1489

No. 1489

PERFORMANCE OF WARM-AIR PERIMETER-LOOP AND PERIMETER-RADIAL SYSTEMS IN A RESIDENCE†

By H. T. GILKEY*, R. W. ROOSE**, AND M. E. CHILDS‡, URBANA, ILL.

THE USE OF the warm-air perimeter type of heating system in basementless houses, especially those with a concrete slab floor, has been accepted in many areas as being one practical answer toward heating these homes satisfactorily. One of the factors which has made this type of warm-air heating system acceptable is that warm floors are provided. Without some heat introduced into the floor of the concrete slab type of home, it has been proved that such difficulties as cold floor surfaces and condensation on these surfaces are likely to occur.[1] Although several variations of the warm-air perimeter system have been used, two of the most popular types are the perimeter-loop and perimeter-radial systems. In the warm-air perimeter-loop system, hereafter referred to as the loop system, the warm air from the furnace is delivered through subfloor feeder ducts to the perimeter duct which is embedded in the floor along the outer walls. The heated air flows through the perimeter duct and enters the rooms through registers usually placed underneath the windows or at other locations where the heat losses are high. In the perimeter-radial system, hereafter referred to as the radial system, the warm air from the furnace is delivered through subfloor feeder ducts directly to the registers and then into the room. In both systems heat is supplied to the rooms by convection from the registers and by a combination of radiation and natural convection from the heated floor surface.

The overall performances of two types of perimeter-loop systems were studied[2] previously in Warm-Air Heating Research Residence No. 3 during the 1949-50 and the 1950-51 heating seasons. In both of these seasons 8-in. diameter ducts were used for both the feeders and the perimeter loop. The performances of three convection systems were also studied in Research Residence No. 3 and compared[2] with the perimeter systems. In addition, laboratory studies of several

---

† This investigation is a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association*. This paper reports one phase of an extensive study conducted in Warm-Air Heating Research Residence No. 3 during the period from 1949 to 1952.

* Research Associate in Mechanical Engineering, University of Illinois. Junior Member of A.S.H.V.E.

** Research Assistant Professor of Mechanical Engineering, University of Illinois. Member of A.S.H.V.E.

‡ Research Associate in Mechanical Engineering, University of Illinois.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

factors affecting the design[3] of perimeter systems using 8-in. diameter ducts were conducted[4] in the Floor Slab Laboratory concurrently with the Residence No. 3 studies.

During the 1951-52 heating season further studies were conducted in Research Residence No. 3 in order to determine the performance characteristics of warm-air perimeter-loop and perimeter-radial heating systems in which 6-in. diameter ducts were used. Each system was studied with two different types of floor registers. One type of register was equipped with deflecting vanes in the register face; the other was not.

The overall objectives of this investigation were to determine for each system studied:

1. Uniformity of room-air temperatures at the sitting level throughout the house, and the response of the system to thermostatic control.

FIG. 1. WARM AIR HEATING RESEARCH RESIDENCE NO. 3

2. Room-air temperature differentials between the floor level and the sitting, breathing, and ceiling levels.
3. Magnitude and pattern of floor-surface temperatures.
4. Panel heating effect of the warmed floor surface.
5. General performance characteristics of the furnace.

DESCRIPTION AND PROCEDURE

Residence No. 3, shown in Fig. 1, was a single story, low-cost home with a concrete slab floor. The Residence, which was of standard construction, was provided with a relatively large amount of glass area. The walls were uninsulated, but the ceiling was insulated with mineral wool batts of 3⅝-in. thickness. With the exception of one picture window, double-hung wood-sash windows were used. The windows were neither weatherstripped nor equipped with storm sash. The outside doors were not weatherstripped, but storm doors were used. A floor plan of the Residence is shown in Fig. 2. The inside dimensions were 24 ft x 32 ft and the corresponding floor area was 768 sq ft. The design heat loss of the Residence for an outdoor temperature of −10 F and an indoor temperature of 70 F was 51,600 Btu per hr.

The floor-slab construction shown in Fig. 3 consisted of a 4-in. fill of coarse graded gravel placed on the original grade, a heavy duplex-paper vapor barrier (dampproofing membrane) and 4 in. of concrete. The insulation at the edge of the slab consisted of a 1-in. asphalt-sheathed glass fiber insulating board placed against the foundation wall and extending downward 12 in. from the top of the slab.

The loop system consisted of a 6-in. diameter warm-air duct embedded in the concrete slab in the form of a single loop around the periphery of the floor with a series of 6-in. diameter feeder ducts connecting the perimeter loop

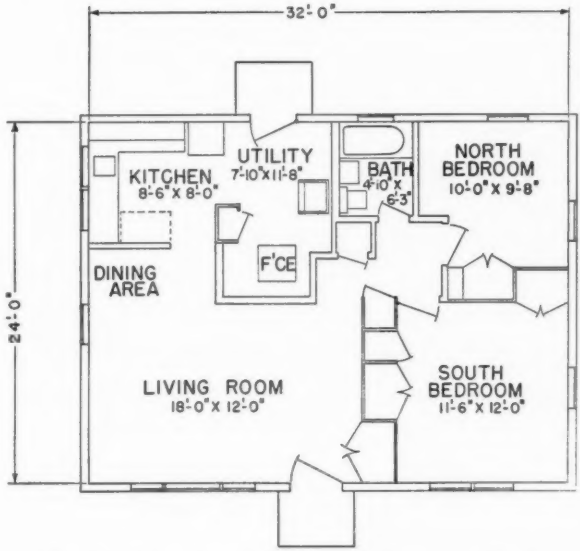


FIG. 2. FLOOR PLAN OF RESEARCH RESIDENCE NO. 3

to a subfloor plenum below the furnace. A total of seven feeder ducts was installed, some of which were equipped with branches. Provisions were made for blocking the feeders at both ends and at the points where the branches joined the feeders so that any combination of the seven feeder ducts could be studied. The top of the feeder ducts was 6 in. below the floor surface at the plenum and sloped upward to the junction with the perimeter loop 2 in. below the floor.

The radial system consisted of six 6-in. diameter feeder ducts which connected the registers to the subfloor plenum. These feeders were the same as those previously discussed. When the radial system was in use, the perimeter loop was sealed on both sides of all registers and where it was joined by feeders not used with the radial system.

A gas-fired down-flow furnace having an input rating of 65,000 Btu per hr for natural gas was used. The warm air was delivered directly into the subfloor plenum below the furnace from which point it was forced outward

through the feeder ducts to the perimeter loop and then through the registers. The air moved across the rooms to a single return-air intake in the living room and into the furnace to be reheated. In the radial system, air was forced outward from the plenum, through the feeder ducts and through the registers.

Two types of registers were used with both the loop and radial systems. One was a 4-in. x 14-in. floor register in which the axis of the shutter damper was parallel to the long axis of the register. The vanes in the face of the register were vertical and thus non-deflecting. The other type was a 2¼-in. x 14-in. floor register in which deflecting vanes were provided. The performance of the two types was compared with identical duct arrangements for both the loop and radial systems.

Five studies were conducted, three loop and two radial, designated as Series P-21, P-22, P-23, R-1, and R-2, respectively, and the duct arrangements are shown in Fig. 4. The four-feeder loop system (Series P-22) was not con-

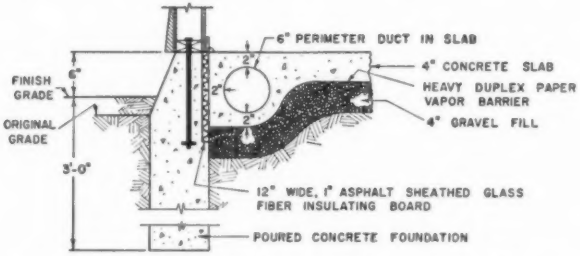


FIG. 3. DETAILS OF FLOOR SLAB CONSTRUCTION SHOWING PERIMETER DUCT

sidered satisfactory because it was not possible to maintain the bedroom-air temperatures at the desired values, but the five-feeder loop systems (Series P-21 and P-23) were considered to be satisfactory as compared[2] with similar systems using 8-in. diameter ducts.

The desired fuel-input rate of 65,000 Btu per hr was obtained by adjusting the burner manifold pressure at the pressure regulator at the furnace. This input rate of 65,000 Btu per hr was determined by dividing the design heat loss of the Residence (51,600 Btu per hr) by the assumed bonnet efficiency (0.80) of the furnace. The thermostat which was located at the 30-in. level in the living room was set at 72 F and this setting was not changed during the heating season. The differential setting of the thermostat was adjusted to the minimum point to provide frequent burner operations. The blower operated continuously, and its speed was adjusted[5] to provide an air temperature rise through the furnace of 100 deg under conditions of continuous burner and blower operation.

Complete instrumentation was provided for temperature measurement, including approximately 250 copper-constantan thermocouples. Instrumentation was also provided to measure the indoor relative humidity, electrical inputs to the gas valve and to the blower motor, the fuel-input rate, the flue gas composition, the hours of operation of the burner, and the air-flow rates in the duct systems.

Since it was not possible to evaluate all of the factors affecting comfort, emphasis was placed upon room-air temperatures and floor-surface temperatures.

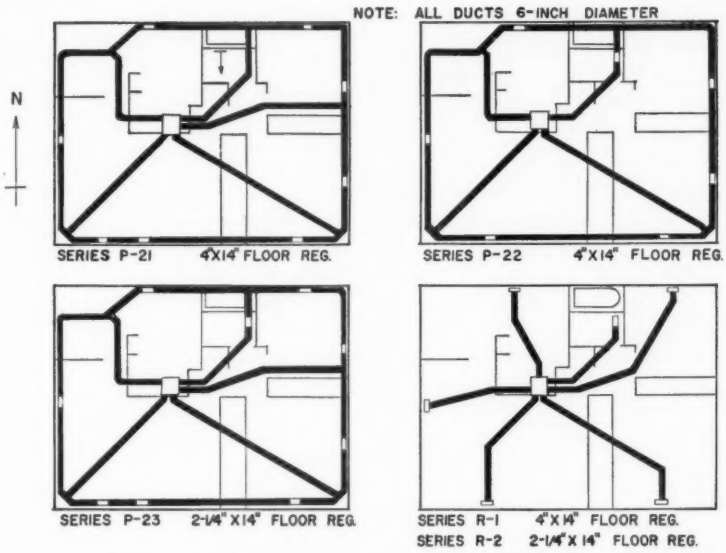

NOTE: ALL DUCTS 6-INCH DIAMETER
N
SERIES P-21 4"X14" FLOOR REG.
SERIES P-22 4"X14" FLOOR REG.
SERIES P-23 2-1/4" X14" FLOOR REG.
SERIES R-1 4"X14" FLOOR REG.
SERIES R-2 2-1/4"X 14" FLOOR REG.

FIG. 4. DUCT ARRANGEMENTS FOR PERIMETER-LOOP AND PERIMETER-RADIAL STUDIES

The study of air temperatures was devoted primarily to the conditions obtained in the living zone between the floor level and the breathing level, since temperatures above the breathing level are of little significance as far as comfort conditions in the home are concerned.

UNIFORMITY OF ROOM-AIR TEMPERATURES AT SITTING LEVEL

The cyclical variation of room-air temperatures was negligible with all five series amounting to less than 0.5 deg. Also, the response of the system to sudden changes in outdoor temperature was considered to be satisfactory since no evidence of lag or overrun of room-air temperatures was experienced during periods of rapidly changing outdoor temperatures.

With the exception of Series P-22, the four-feeder loop system, little difficulty was encountered in balancing the systems. By adjusting the dampers in the floor registers, the air temperature in any room could be readily raised or lowered. The best balance was obtained with Series P-21 and P-23, the five-feeder loop systems. The temperature difference between the living room and the two bedrooms was consistently less than 1 deg, whereas with the two radial systems the temperatures in the bedrooms were generally 1 to 2 deg lower than those in the living room.

The temperature difference between the warmest and coolest areas of the living room was less than 2 deg for all four series of studies. When the loop systems were in use and the feeder duct extended into the exposed corner of

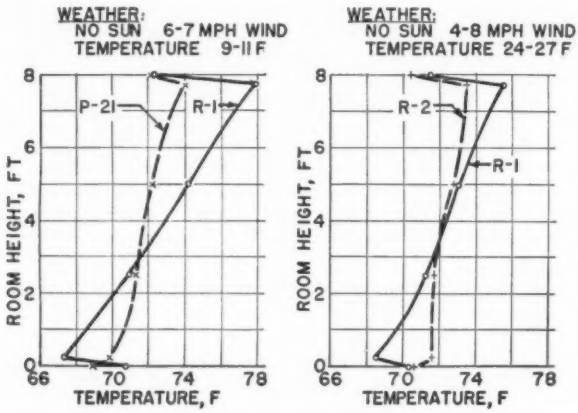


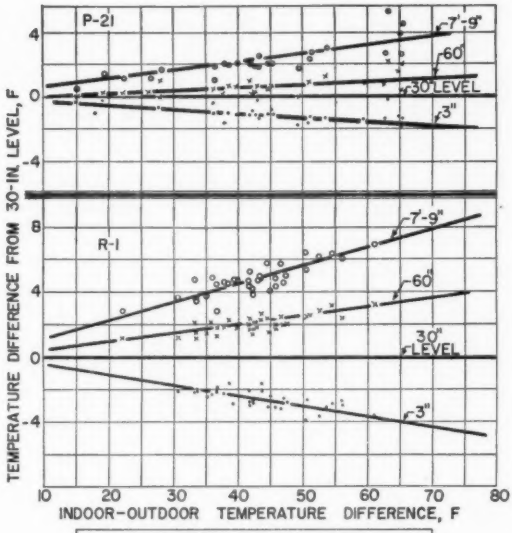
FIG. 5. ROOM-AIR TEMPERATURE GRADIENTS IN NORTH BEDROOM

the living room, the air temperatures in the corner were slightly higher than when the radial systems were in use. They were also higher for a given duct arrangement when deflecting registers were used than they were with straight-vane registers in use (Series P-23 *vs.* P-21 and R-2 *vs.* R-1). This confirms the desirability of extending the feeder ducts into the exposed corners as reported from the previous studies[2] and indicates the desirability of introducing the air into the room so that it is more evenly distributed along the exposed walls.

ROOM-AIR TEMPERATURE DIFFERENTIALS

Satisfactory temperature differentials were obtained with both of the loop and both of the radial systems. The temperature gradients experienced in the north bedroom are shown in Fig. 5, and are representative of those experienced in the other rooms of the Residence. The comparison of Series P-21 and R-1 is shown for an outdoor temperature of about 10 F and a 6 to 7 mph wind. The more nearly vertical gradient experienced with Series P-21 has been attributed to the panel heating effect of the warmed floor. As will be discussed in the sections on Floor-Surface Temperatures and Heat Emission from the Floor, the warmer floors experienced with the loop system caused a greater panel heating effect and a consequent reduction in the amount of heat admitted into the rooms through the registers. For the same air-flow rate, the register-air temperatures were lower for the loop system, and as a result the air temperatures at the ceiling and breathing levels were also lower. In Fig. 5 it should be noted that the floor-surface temperature was higher for the radial system than for the loop system due to the proximity of the feeder to the floor-surface thermocouple. Since these temperatures were measured at the same locations for all series, changing to the feeder duct for the radial system affected the floor-surface temperature, but not the air temperatures.

The effect of the deflecting type of floor register is shown in the comparison of the gradients for Series R-1 (non-deflecting) and R-2 (deflecting). This



TEMPERATURE DIFFERENTIALS, F								
	INDOOR-OUTDOOR DIFFERENCE							
	35F				70F			
	P-21	P-23	R-1	R-2	P-21	P-23	R-1	R-2
3'FL - 30"FL	0.8	1.2	2.1	1.0	1.7	2.0	4.2	1.0
3'FL - 60"FL	1.4	1.4	3.7	2.6	3.0	2.4	7.8	2.3
3'FL - 3"CLG	2.8	2.3	5.9	3.2	5.5	3.3	12.0	3.3

FIG. 6. ROOM-AIR TEMPERATURE DIFFERENTIALS IN NORTH BEDROOM

comparison was made at an outdoor temperature of approximately 25 F and a 4 to 8 mph wind. The deflecting vanes in the register face caused better mixing of the air leaving the register with the air in the room which in turn improved the room-air temperature gradients. When the deflecting register was used with the loop system (Series P-23), its effect was similar though not so pronounced as when it was used with the radial system.

The effect of outdoor temperature on the room-air temperature differentials in the north bedroom is shown in Fig. 6 for Series P-21 and R-1. These differentials are typical of those experienced in other rooms of the Residence. The heated floor panel produced by the loop system decreased the effect of outdoor temperature on the differentials from that experienced with the radial system. The differentials for indoor-outdoor temperature differences of 35 and 70 deg are tabulated in Fig. 6 for all four series. It can be seen that the floor to ceilings differentials were more satisfactory when the deflecting register was used. The effect of the deflecting type register was much more pronounced when used with the radial system than when used with the loop system.

The same bathroom register location was not used for all series. For Series P-21, air from the perimeter loop was admitted underneath the bathtub. The air then moved into the room through a long narrow grille located at the base

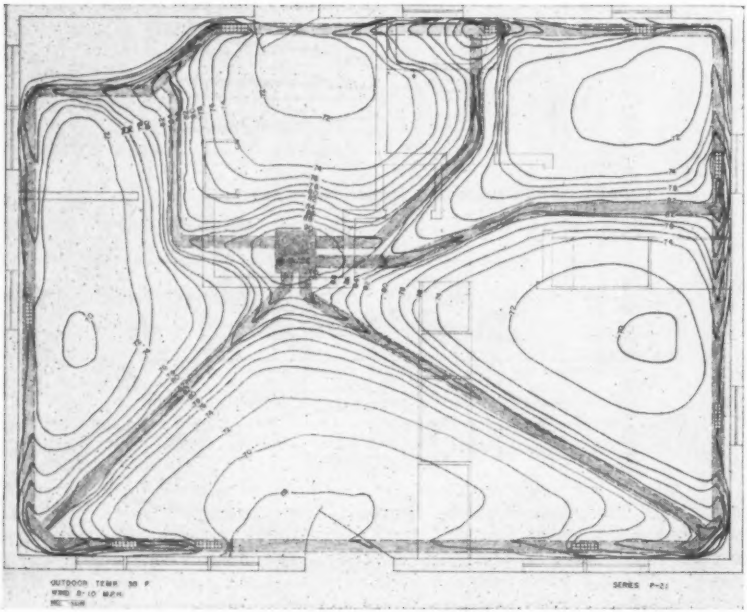


FIG. 7. FLOOR-SURFACE ISOTHERMS WITH PERIMETER-LOOP SYSTEM (SERIES P-21)

of the tub. For Series P-23, a 2¼-in. x 14-in. deflecting floor register was placed directly over the feeder duct as shown in Fig. 4. At an outdoor temperature of 35 F, the floor to breathing level differential was 1.5 deg for Series P-21 and 4.2 deg for Series P-23. The floor to breathing level differentials in the bathroom were 4.4 deg for Series R-1 and 3.6 deg for Series R-2. This would seem to indicate that it is not advisable to place a register so that it delivers air directly from a feeder duct in a perimeter-loop system.

FLOOR-SURFACE TEMPERATURES

The patterns of floor-surface temperatures obtained with the loop system (Series P-21) were more satisfactory than those obtained with the radial system (Series R-1). Typical floor-surface isotherms for Series P-21 are shown in Fig. 7. These isotherms show the conditions at the end of a three day period during which the outdoor temperature was between 30 and 35 F and the sky was overcast. Fig. 8 shows similar isotherms for Series R-1.

When the loop system was in use, the temperatures of only small areas of the floor surface were less than 70 F. The largest of these areas was in the living room near the front door where the floor-surface temperature was less than 68 F near the edge of the slab. Another low temperature area was in the kitchen where the perimeter loop did not extend under the built-in cabinets.

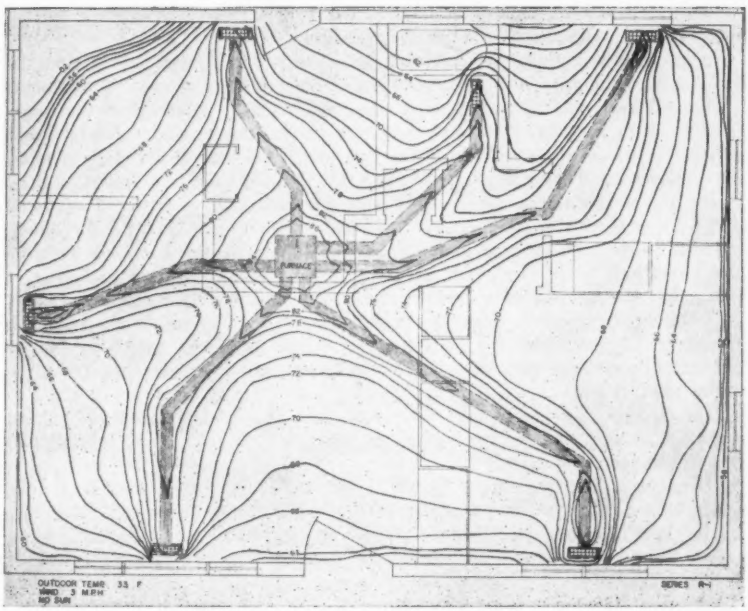


FIG. 8. FLOOR-SURFACE ISOTHERMS WITH PERIMETER-RADIAL SYSTEM (SERIES R-1)

The other areas with floor-surface temperatures below 70 F were in the west portion of the living room and in the south bedroom. A limited area having floor-surface temperatures in excess of 90 F existed in the immediate vicinity of the furnace and subfloor plenum. However, this was not considered to be objectionable since the areas were small and were not used as living areas.

Although the areas of low floor-surface temperatures were relatively small and were located where they were not likely to be objectionable, some minor improvement could be made by suitable balancing of the system. This could be done by increasing the air-flow rate in the sections of the perimeter loop near the cold areas. It is possible that this might disturb the room-air temperature balance in the Residence by lowering the air delivery from some registers and increasing it from others. The most effective means of improving the pattern of floor-surface temperatures, however, would be to locate additional feeders leading to the low temperature areas.

Areas of low floor-surface temperature were considerably greater with the radial system (Fig. 8) than they were with the loop system. Near the exposed edge of the slab, the floor-surface temperature was greater than 70 F only in the immediate vicinity of the feeder ducts. Between feeders the floor-surface temperature was below 60 F at the edge of the slab, and areas having surface temperatures less than 70 F extended well in toward the center of the Residence.

It would of course be impossible to alter the floor-surface temperature pattern appreciably by rebalancing the perimeter-radial system.

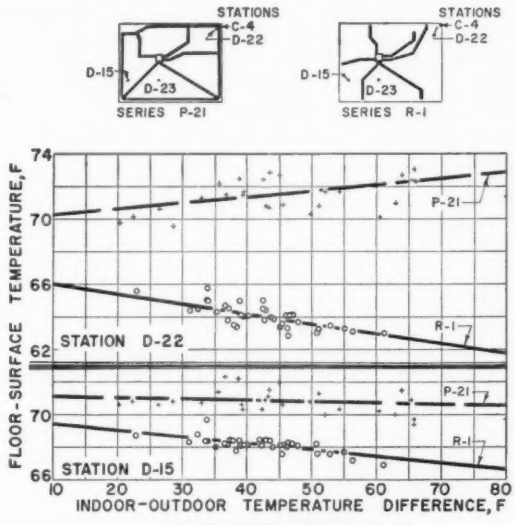
Previous investigations[2] in Research Residence No. 3 have shown that the floor-surface temperatures experienced with a convection system with the registers located high on the inside walls were considerably lower than those experienced with the radial system. Although the minimum floor-surface temperatures were not appreciably different for the high wall and radial systems, with outdoor conditions similar to those for Figs. 7 and 8 the entire floor surface had temperatures below 70 F with the high-wall system and approximately one-half of the floor surface had temperatures less than 65 F. Along with these low floor-surface temperatures, less satisfactory room-air temperature conditions were experienced.

Although the floor-surface temperatures were lower with the radial system than with the loop system, the effect of outdoor temperature on the floor-surface temperatures was similar for both systems except near the periphery of the residence. For the purpose of this analysis the floor area of the test residence can be divided into three general regions: (1) directly over or in the immediate vicinity of feeder ducts; (2) away from the influence of feeder ducts, the perimeter loop, and the exposed edge of the slab; and (3) near the exposed edge of the slab.

In Region 1 directly over or in the immediate vicinity of feeder ducts, the surface temperatures increased as the outdoor temperature decreased. This was true for both the loop and the radial systems and was caused by the higher duct-air temperatures necessary to compensate for the increased heat loss of the Residence.

In Region 2 away from the influences of feeder ducts, the perimeter loop, and the exposed edge of the slab, the floor-surface temperature decreased with decreasing outdoor temperature. This is shown for Series P-21 and R-1 by the curves labelled D-15 in Fig. 9. Although the presence of the perimeter loop in Series P-21 undoubtedly caused the surface temperature to be higher at this point than it was when the loop was not in use (Series R-1), the slopes of the curves are typical of those encountered for Region 2. It can be seen that the outdoor temperature had an only slightly greater effect on the floor-surface temperature with the radial system than with the loop system since the slope of the curve is slightly greater for Series R-1 than for Series P-21. This is in agreement with the results[2] found in Residence No. 3 during the 1949-50 heating season when the high-sidewall and low-sidewall convection systems were in use.

In Region 3, near the exposed edge of the slab the outdoor temperature effect on the loop system was quite different from the effect on the radial system. This comparison for Series P-21 and R-1 is shown by the curves labelled D-22 in Fig. 9. At this point, approximately 1.5 ft from the north and east walls of the Residence, the floor-surface temperature increased as the outdoor temperature decreased when the perimeter loop was in use (Series P-21). It decreased sharply with decreasing outdoor temperature, however, when the perimeter loop was not in use (Series R-1). The curve for Series P-21 is representative of the temperatures observed on the floor surface in the vicinity of those sections of the perimeter loop in which there was sufficient air flowing. Near those sections of the perimeter loop in which there was little air motion, such as along the south exposure near the front door, the floor-surface temperatures for Series P-21 were similar to those away from both feeder and



FLOOR-SURFACE TEMPERATURE, F				
	INDOOR-OUTDOOR DIFF.			
	35 F		70 F	
	P-21	R-1	P-21	R-1
D-15	70.9	68.4	70.5	67.0
D-22	71.1	64.4	72.4	62.4
C-4	71.9	61.9	73.1	56.5
D-23	70.5	69.7	69.7	67.9

FIG. 9. EFFECT OF OUTDOOR TEMPERATURE ON SELECTED FLOOR-SURFACE TEMPERATURES

perimeter ducts *i.e.*, higher and with less effect from the outdoor temperature than for Series R-1. The curve for Series R-1 is representative of the temperatures observed on the floor surface near the exposed edge of the slab for the radial system.

HEAT EMISSION FROM THE FLOOR

Although both the loop and radial systems are primarily convection heating systems, there is an appreciable panel-heating effect from the warm floor. This is particularly true of the loop system in which most of the floor surface is at a temperature greater than that of the air in the rooms. This heat emission from the floor can be considered as having two components: radiation heat transfer from the floor surface to the walls and ceiling, and heat transferred by natural convection from the floor surface to the air adjacent to it. The methods used to calculate the total heat emission are the same as used by Sjordal[6] and Sadler[7] in the special studies that have been conducted and reported[2] for Residence No. 3.

The radiation heat transfer depends upon the area and temperature of the emitting surface, the temperature of the receiving surface, and the configuration

factor between the emitting and receiving surfaces. The equation recommended by McAdams,[8] which is based on the Stefan-Boltzmann law, was used:

$$Q_r = 0.173 \times 10^8 A_S F_A (T_S^4 - T_R^4) \quad \ldots\ldots\ldots (1)$$

where

Q_r = radiation heat transfer, Btu per (hour) (square foot).
A_S = surface area of the emitting surface, square feet.
T_S = absolute temperature of the emitting surface, Fahrenheit, absolute.
T_R = absolute temperature of the receiving surface, Fahrenheit, absolute.
F_A = configuration factor which considers the solid angle between the source and the receiver, the areas, and the emissivities of the source and the receiver.

The configuration factor can be evaluated by:

$$F_A = \frac{1}{\frac{1}{F_b} + \left[\frac{1}{e_S} - 1\right] + \frac{A_S}{A_R}\left[\frac{1}{e_R} - 1\right]} \quad \ldots\ldots\ldots (2)$$

where

F_b = configuration factor for black body radiation.
e_S = emissivity of the emitting surface.
e_R = emissivity of the receiving surface.
A_S = area of the emitting surface, square feet.
A_R = area of the receiving surface, square feet.

Since the emitting surface, the floor, was enclosed on all sides by the receiving surfaces, the walls and ceiling, both F_b and A_S/A_R can be considered as equal to 1. The emissivities of both the floors and the walls and ceilings were assumed to be 0.90. The value of the configuration factor F_A was then

$$F_A = \frac{1}{1 + \left[\frac{1}{0.9} - 1\right] + 1\left[\frac{1}{0.9} - 1\right]} = 0.82$$

The area of the emitting surface was measured for each 2 deg increment of floor-surface temperature T_S. For instance, for Series P-21, the area between two adjacent isotherms was measured from Fig. 7, and the temperature was taken as the average of those represented by the isothermal lines. Thus, the area between the 80 F and 82 F isotherm was found to be 24.12 sq ft and this was assumed to be at a temperature of 81 F or 541 R (F absolute).

The temperature of the receiving surface, T_R, was the weighted average temperature of the walls, ceiling, and windows of the Residence and was evaluated by the average surface temperature method. Thus, the average surface temperature for any room, T_R, was the sum of the products of the wall, ceiling, and window areas and temperatures divided by the total area of walls, ceiling, and windows of the room. This may be expressed as:

$$T_R = \frac{(A_N T_N + A_S T_S + A_E T_E + A_W T_W + A_C T_C + A_G T_G)}{(A_N + A_S + A_E + A_W + A_C + A_G)} \quad \ldots\ldots (3)$$

The subscripts N, S, E, and W refer to north, south, east and west walls, C refers to ceiling, and G refers to windows. Although this average surface temperature method lacks the precision of the mean radiant temperature method, the computation of configuration factors for use with the latter method would

be extremely complex and tedious due to the large window areas and to the wide range and non-uniform pattern of floor-surface temperatures. Furthermore, Jamieson,[4] has shown that little error is introduced by the use of the less precise average surface temperature method.

The natural convection heat transfer from the floor surface to the air adjacent to it was calculated using King's[9] equation which was found by Jamieson to give good results. For sections of the floor warmer than the adjacent air, the convection component of the total heat emission from the floor surface becomes:

$$Q_c = 0.28\, A_S \theta^{1.33} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

where

Q_c = heat transfer by natural convection, Btu per (hour) (square foot).
θ = temperature difference between the surface and the air, Fahrenheit degrees.

For sections of the floor colder than the adjacent air

$$Q_c = 0.15\, A_S \theta^{1.33} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

In both cases A_S was evaluated in the same manner as for the radiation component, and θ was evaluated from t_S, the average floor-surface temperature of A_S, and the average air temperature 3 in. above the floor. In the case of the loop system, Series P-21, this average air temperature was 70 F; for the radial system, Series R-1, it was 68 F.

Table 1 shows the total heat supplied to each room of the residence for Series P-21 and R-1. For the loop system the heat emitted from the floor, 5,260 Btu per hr, was 44 percent of the total energy, 11,990 Btu per hr, which entered the rooms from the air heated in the furnace. In the case of the radial system, the heat emitted from the floor, 1,950 Btu per hr, was 16 percent of the total energy, 12,070 Btu per hr, which entered the rooms. For a three-feeder loop system using 8-in. diameter ducts this value was found[2] to be 3,100 Btu per hr, or 25 percent of the total when the outdoor conditions were essentially the same as for Series P-21 and R-1.

Although a complete energy balance study was not made for either Series P-21 or R-1, the excellent agreement between the total heat supplied to the rooms for the two systems indicates that the results are reasonable. Sources of energy which were not considered in these studies were heat gains from the furnace casing and flue pipe, energy from observers, and miscellaneous electrical inputs such as lights, controls, etc.

PERFORMANCE OF FURNACE

Complete performance data for the furnace were obtained over a wide range of indoor-outdoor temperature differences. The general performance characteristics were essentially the same for all four series since the operating conditions were identical. These conditions were in accordance with the continuous air circulation principle,[5] with the one exception mentioned previously. The differential setting of the room thermostat was adjusted to provide short cycling of the burner during mild weather, the fuel-input rate was adjusted to equal the heat loss of the house divided by the assumed bonnet efficiency, and the blower was adjusted to provide a temperature rise through the furnace of 100 deg.

A summary of the performance data for Series P-21 for an outdoor temperature of 35 F is shown in Table 2.

Comparisons of the fuel consumptions of the four systems are given in Table 3. The fuel consumption was about 16 percent less for the radial system than for the loop system. This was attributed primarily to the larger subfloor and edge losses which were experienced when the perimeter loop was in use. It should be noted, however, that the fuel consumption of the loop system was less than that which was experienced with a convection system[2] using high-sidewall delivery.

PRESSURE LOSSES FOR THE DUCT SYSTEMS

In connection with the blower performance, measurements were made of the pressure losses for the duct systems for Series P-21, P-23, R-1, and R-2. These measurements, which are shown in Table 4, were made on mild days when the furnace had been off for some time and the air temperature in the subfloor plenum was approximately 80 F. During periods of furnace operation, the pressure loss of the warm-air duct system remained constant and the pressure loss of the return-air system decreased. This occurred because the blower was located below the heat exchanger, and since the blower delivered an essentially constant volume of air, the air-flow rate in the warm-air ducts remained constant as the air temperature in the subfloor plenum increased. In the return-air duct, however, the air-flow rate decreased and the return-air temperature remained constant as the air temperature in the subfloor plenum increased.

For Series P-21, the five feeder loop system, the pressure loss was 0.08 in. of water for an air-flow rate of 430 cfm. With the same duct system but with the deflecting type of register in use, Series P-23, the pressure loss was 0.11 in. of water. This increase was attributed solely to the use of the smaller deflecting register which imposed a greater resistance to air flow upon the blower. A similar difference was experienced with the radial systems, Series R-1 and R-2.

The pressure loss on the warm-air side of the system was found to be greater with the radial systems than with the loop systems. This difference was attributed

TABLE 1—HEAT EMITTED FROM FLOOR AND THROUGH REGISTERS FOR PERIMETER-LOOP AND PERIMETER-RADIAL HEATING SYSTEMS

Room	Perimeter-Loop System					Perimeter-Radial System				
	Heat Emission from Floor, Btu/hr			Heat from Reg Btu/hr	Total to Room, Btu/hr	Heat Emission from Floor, Btu/hr			Heat from Reg Btu/hr	Total to Room, Btu/hr
	Radiation	Convection	Total			Radiation	Convection	Total		
Liv Rm....	1,260	630	1,890	1,940	3,830	440	440	880	3,020	3,900
S Bed Rm..	650	300	950	1,590	2,540	—70	70	0	1,770	1,770
N Bed Rm.	420	230	650	1,800	2,450	—10	80	70	1,830	1,900
Hall.....	190	210	400	—	400	110	170	280	—	280
Bathroom..	120	100	220	320	540	—20	30	10	1,260	1,270
Kitchen-Utility.....	680	470	1,150	1,080	2,230	320	390	710	2,240	2,950
Total....	3,320	1,940	5,260	6,730	11,990	770	1,180	1,950	10,120	12,070

Weather: Series P-21: Outdoor Temp 35 F; Wind 7.5 mph (SSE); Sky Overcast.
Series R-1: Outdoor Temp 36 F; Wind 3 mph (SSE); Sky Overcast.

TABLE 2—SUMMARY OF PERFORMANCE DATA FOR BURNER AND BLOWER FOR SERIES P-21 AT AN OUTDOOR TEMPERATURE OF 35 F

a. Blower Operation, hr per day	24
b. Electrical Input to Blower Motor, Watt-hr per day	3,100
c. Average Bonnet-Air Temperature, F	110
d. Burner Operations, hr per day	8
e. Burner Operations, per 24 hr	167
f. Average Flue-Gas Temperature, F	220
g. Fuel Consumption, cu ft per day	470

to several factors. Although six feeder ducts were used with the radial system, one more than with the loop system, the feeder to the bathroom carried very little air. In addition the feeders for the radial system had more turns and were longer in some cases than those used with the loop system. Furthermore, fewer registers were used thus increasing the resistance in each register. It should be noted, however, that only in the case of Series R-2 did the pressure loss for the entire system equal 0.20 in. of water, the value usually used for designing warm-air heating systems.

CONCLUSIONS

The warm-air perimeter heating systems discussed in this paper were of two types. One was a five-feeder perimeter-loop system; the other was a six-feeder perimeter-radial system. The studies involved the use of conventional floor registers with straight vanes in the face and floor registers with deflecting vanes in the face with both duct systems. The ducts were of 6-in. diameter and were embedded in the concrete floor of the Residence. The same down-flow furnace was used with each system, and the warm air was forced directly from the furnace into the subfloor plenum and thence into the feeder ducts. In the case of the radial systems, the feeder ducts carried the air directly to the floor registers. In the case of the loop systems, the air flowed through the feeder ducts to the perimeter loop and through it to the registers. After entering the rooms through the registers, the air moved to a single return-air intake in the living room and then into the furnace to be reheated.

The response of the system to sudden changes in outdoor temperatures was considered to be satisfactory since no evidence of lag or overrun of room-air temperatures was experienced during periods of rapidly changing outdoor temperatures. The conventional room thermostat used with the systems proved to be satisfactory for controlling the room-air temperature. The practice of bal-

TABLE 3—COMPARISON OF FUEL CONSUMPTIONS OF PERIMETER-LOOP AND PERIMETER-RADIAL SYSTEMS

SERIES	FUEL CONSUMPTION CU FT PER DAY	
Outdoor Temp.,	35	0
P-21	470	940
P-23	460	910
R-1	400	750
R-2	390	780

ancing the system by means of adjusting the shutter dampers on the floor registers proved to be satisfactory for both the loop and the radial systems.

Room-air temperature differentials were small. For example, at an outdoor temperature of 20F, the temperature differential from the floor level to the breathing level was 2.1 deg for the loop system and 5.5 deg for the radial system. When the deflecting registers were used, these differentials were 1.8 and 2.9 deg for the loop and radial systems, respectively.

The floor-surface temperature patterns experienced with the radial system were not as satisfactory as those experienced with the loop system. With the loop system only three small areas were observed in which the floor-surface temperatures were less than 70 F when the outdoor temperature was 35 F. Large areas of the floor with temperatures less than 70 F were observed with the radial system under similar weather conditions, and near the exposed edge of the slab the floor-surface temperatures were less than 65 F.

The heat emitted from the floor surface for the loop system was 44 percent of the total heat which entered the rooms from the air heated in the furnace. For the radial system, 16 percent of the total heat which entered the rooms was emitted from the floor surface. Both of these values were calculated for an outdoor temperature of 35 F.

The fuel consumption with the radial system was about 16 percent less than with the loop system, and both were less than that previously experienced with a high-sidewall convection system. The pressure loss of the duct system was less with the loop system than with the radial system. The use of the small deflecting floor registers increased the pressure loss slightly, but it did not exceed 0.20 in. of water with any of the duct systems studied.

This is a report of studies conducted with the duct systems shown in Fig. 4. The radial systems were not designed in accordance with the recommendations[10] contained in the latest edition of *Manual 4* of the *National Warm Air Heating and Air Conditioning Association.* Had additional feeders and registers been incorporated into these systems, the room-air temperature balance and the floor-surface temperature patterns would have been improved. In addition, the pressure losses in the duct systems would have been smaller.

ACKNOWLEDGMENT

This paper reports one phase of a comprehensive investigation conducted in Warm-Air Heating Research Residence No. 3 under the terms of a cooperative agreement between the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station of the University of Illinois.

TABLE 4—PRESSURE LOSSES FOR PERIMETER-LOOP AND PERIMETER-RADIAL SYSTEMS

SERIES	PRESSURE LOSS, IN. OF WATER			AIR-FLOW RATE, cfm
	Warm-Air	Return-Air	Total	
P-21	0.08	0.05	0.13	430
P-23	0.11	0.05	0.16	440
R-1	0.12	0.05	0.17	450
R-2	0.16	0.04	0.20	435

These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is hereby made to the manufacturers who cooperated by furnishing materials and equipment used in this investigation.

REFERENCES

1. Measurement of Heat Losses from Slab Floors, by R. S. Dill, W. C. Robinson, and H. E. Robinson (*National Bureau of Standards Building Materials Report BMS 103,* 1945).

2. Comparative Performances of Two Warm-Air Perimeter Systems and Three Convection Systems, by M. E. Childs, R. W. Roose, H. T. Gilkey, and S. Konzo (*University of Illinois Engineering Experiment Station Bulletin 403,* 1952).

3. *Manual 4,* Warm Air Perimeter Heating (2nd edition, *National Warm Air Heating and Air Conditioning Association,* Cleveland, 1951).

4. Heat Emission Characteristics of Warm-Air Perimeter Heating Ducts, by J. R. Jamieson, R. W. Roose, H. T. Gilkey, and S. Konzo (*University of Illinois Engineering Experiment Station Bulletin 412,* in press).

5. *Manual 6,* Service Manual for Continuous Air Circulation Technicians (First Edition, *National Warm Air Heating and Air Conditioning Association,* Cleveland, 1947).

6. Heat Balance of Residence Heated by Perimeter Duct System, by J. E. Sjordal (Unpublished Thesis, University of Illinois Library, 1950).

7. Heat Balance of a Radial-Feeder Perimeter Heating System, by G. W. Sadler (Unpublished Thesis, University of Illinois Library, 1951).

8. *Heat Transmission,* by W. H. McAdams (McGraw-Hill Book Co., New York, Second Edition, 1942).

9. The Basic Laws and Data of Heat Transmission, by W. J. King (*Mechanical Engineering,* Vol. 54, May 1932).

10. *Manual 4,* Warm Air Perimeter Heating as Applied to Structures Without Basements (3rd Edition, *National Warm Air Heating and Air Conditioning Association,* Cleveland, 1953).

DISCUSSION

C. M. TOELAER, Bloomfield, N. J. (WRITTEN): This paper is a thorough study of a perimeter-loop system but I feel that the radial system has been presented in a very unfavorable way. The tests on the radial system were not made on a properly installed system as pointed out in the last paragraph of the paper.

The authors, I believe, will agree that the degree of comfort and the stratification in the space are a function of the number of outlets installed, not on the number of buried feeder ducts, and that the floor surface temperature and the uniformity of surface temperature is a function of the number of feeders. A combination of minimum stratification and warm floors is desirable.

On this basis then, if tests were conducted on a radial system with the same number of outlets as were used in the loop system, stratification would be a function of register design and leaving air velocity, and would be equal to or better than the loop system which does not offer quite as positive and effective means of balancing. The floor surface temperature would thus be more uniform than with the loop system because the number of radials would be increased.

The isotherm lines in Fig. 7 indicate clearly that there is virtually no air movement in the perimeter duct at the front of the house between the floor registers. This was shown much better and for all sides of the house in the previous paper* by the authors. A system with 3 or 4 feeders is rather common practice in the industry.

* Performance of a Warm-Air Radial-Feeder Perimeter-Loop System in a Residence, by M. E. Childs, R. W. Roose and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 235).

If the loop duct is carrying no, or virtually no air, isotherm lines should be virtually unchanged if the duct were omitted. Basically, then, a perimeter-loop system is a radial system with two registers connected by branch ducts to the single feeder.

All of us recognize that steam flowing from two boilers into a bull head tee results in a bouncing water line. Similarly a bull head smoke pipe tee connection on two oil-fired furnaces or boilers results in flame pulsation. Air is also a gas. Why is it then that we expect good air distribution control when air is brought into a duct from two directions?

In homes for the modest income group floor space is at a premium and every possible foot should be made available to the home owner. In small compact homes, floor space costs a home owner about $12.00 per sq ft. The loss of 6 to 8 sq ft of usable space for each outlet because furniture cannot be placed near floor registers, will amount to over $800.00 in a house so equipped. Register design is of utmost importance in basementless homes to avoid window down drafts and floor drafts, but consideration must also be given to freedom of furniture placement and use of maximum space.

I think the tests have been excellent and should be continued. The industry needs more such comprehensive studies. Unfortunately, the tests compare one of the best methods of doing the job with the loop system with a radial system which is not up to the recommended standards. The best of both or trade practice in both should be the basis of the comparison. It might be well to conduct tests on a radial system as currently recommended by *National Warm Air Heating and Air Conditioning Association* with a loop system feeding the four corners.

Even with a radial system which is not in complete conformity with current recommendations, the paper showed 16 percent lower operating cost. Also, even with fewer outlets, the radial system, according to Fig. 6, had a room temperature gradient, under the most severe weather conditions, equal to, or better than, the loop system. It might have been better, had a more favorable radial system design been tested.

W. M. MYLER, JR., Columbus, Ohio (WRITTEN): This paper reports a new phase in the series of tests which have been run to determine the fundamentals involved in satisfactorily heating a typical one-story house built on a concrete slab floor.

Previously, we have been given data on a gravity system and forced-air systems with high and low wall type of registers. Papers on perimeter-loop systems made with 8-in. ducts have also been presented. This paper carries us one step farther in showing results for a 6-in. duct perimeter-loop with radial feeders as well as for a radial-perimeter system without loops in which the registers are installed directly at the end of the radial feeders.

A study of not only the data contained herein but also those previously published on this subject now permit one to make an evaluation of the heating results obtainable, the cost of installation, and the relative operating costs for a number of possible types of warm-air systems which could be applied to this concrete floor slab home.

Gravity systems and forced-air systems using either high wall or low wall registers failed to produce satisfactory floor slab temperatures even when the slab was properly edge insulated.

Either 8-in. or 6-in. pipe can be used to make a highly satisfactory perimeter-loop radial feeder system, but the operating cost is appreciably greater than that of a simple radial-perimeter system with registers installed directly at the end of the radials. The latter system would appear to be both cheaper to install and operate than the former and yet its performance is superior to a gravity or a conventional forced air system in this type of home.

While no series of tests can ever be said to be complete, at least this one has progressed to the point where the pitfalls of certain systems have become obvious. When these are avoided there still remains a reasonable range of choice of system

design which will give generally satisfactory performance and yet which may vary considerably in first cost and in operating cost depending upon the individual desires.

R. K. THULMAN, Washington, D. C.: The distinguishing characteristic of the many fine papers describing the cooperative projects of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association* is the facility with which the results can be interpreted and put into actual practice. This paper, which is no exception, has added considerably to the store of much-needed information on the heating of small houses and underlines the importance of a more generous and judicious use of insulation.

The house in which the tests were conducted seems to have been very poorly insulated. Its heat loss is higher even than the 60 Btu per sq ft limit permitted under the F.H.A. Minimum Property Requirements. The edge insulation falls short of the recommendation of the Small Homes Council, the *N.W.A.H.A.C.A.* and the F.H.A. There was no insulation in the walls and the windows were not weather stripped.

It is concluded, therefore, that the difference of 16 percent between the operating costs of the two systems, the loop system showing the higher operating cost, would have been considerably less had the structure been insulated more effectively, even if it had been insulated in accordance with minimum requirements.

It is noted that the calculated heat loss was 51,000 Btu per hour while a calculation based on the fuel consumption results in an estimated heat loss of only 40,000. It would be interesting to account for this difference.

Since the edge insulation was so poor it is conceivable that additional insulation in the walls above the slab edge might have reduced the operating costs of the loop system bringing them more nearly in line with those of the radial system.

R. S. DILL, Washington, D. C.: This paper, of course, is another example of the thorough kind of work that comes from the University of Illinois. If it has a fault, it does not describe in enough detail what the weather was like during these tests, because the hygrometric state of the earth under these heating ducts has a lot to do with the performance of the system. If there is rain or snow causing puddles next to the outside wall of the house, there is going to be a hydrostatic head tending to force water into the ducts. It is unreasonable to expect either a concrete wall or a floor or one of these so-called vapor barriers to be water-tight, so it is presumable that under this condition water will enter the duct. In winter when the heat is on, some water will come out as vapor and make the humidity in the house excessive. I have seen puddles in ducts under houses, and it is obvious the water will cause excessive humidity in the house.

The plan presented showing the arrangement of the floor indicated the bottom of the ducts to be 2 or 3 in. below the outside grade. If it were my house, I would insist that the duct be above the outside grade. The tests described in the paper demonstrate that you can heat a house, and satisfactorily heat it, with either of these systems under good conditions. As far as we know there was no rain or snow during these tests. It being laboratory equipment presumably it was all in very good order. But I believe some experiments are in order when the conditions are bad, when there is rain and snow and slush on the ground outside and when there are defects in the workmanship.

There are other indications that this is a type of heating system that requires thorough supervision on the job. I know a case in which some clay tile pipe was used for ducts, and the warm air escaped through the foundation wall of the building.

D. L. OSLUND, Minneapolis, Minn.: In the City of Minneapolis we have had considerable experience with perimeter heating in the last 14 months. Much pressure has been applied to the Department of Buildings to allow the use of slab perimeter heating. Several installations were made within the city. At first, everything worked

well until some contractors began to cut down on supervision, allowing the general contractor and cement men to install the insulation along the perimeter of the slab. This condition became unsatisfactory to the Department of Buildings and as a result we refused to allow any future installations of perimeter heating within the City.

We understand that the thought behind the design of perimeter heating was to reduce the cost of homes in the lower priced field. It is somewhat of a mystery why the heating system should be singled out to cut building costs when such system is about 10 percent of the construction costs. There is no more important part of any home than its heating system, especially in Minneapolis.

The use of floor registers seems a backward step. These floor registers not only prevent placement of furniture, require cutting holes in carpets, but become dust collectors when not in use; and when in use, continually fill the air stream with dust which is spread all over the house.

Why should we put filters in a warm-air heating unit to collect the dust being distributed in the home and then use dust spreading floor registers? We feel that this method of distribution will set the warm-air heating industry back many years.

J. L. ANTONIO†, Denver, Colo.: What effect will moisture in concrete have on fiber ducts when the ducts are completely encased in concrete? In Denver we have not yet adopted any ordinance or code governing perimeter loop or radial heating systems. We feel that the theory embodied in such systems requires additional studies. We have not followed the lead of Minneapolis.

W. W. GREILING, Cleveland, Ohio: I think it is very poor policy for a public official to take such an attitude as Mr. Oslund took about banning certain types of heating systems. Even though it might not be considered by some people as a good heating system, anybody who wants that system is entitled to spend his money for it. The American system of free enterprise implies that any man who wants to spend his money may spend it for what he wants. That is the accepted practice of free men in a democratic country, and it has resulted in progress and a high standard of living for all of us. The fact that several installations of this system were not adequately designed or correctly installed is no reason for some city official to condemn all perimeter systems and to advocate that they shall not be installed.

P. W. YOUNG, Denver, Colo.: I would like a little discussion on different types of tubes to be used in the system such as fiber, concrete, tile, clay tile and metal.

AUTHORS' CLOSURE (H. T. Gilkey): It has been pointed out by Mr. Thulman that the heat loss of Research Residence No. 3 is greater than the maximum permitted under the F.H.A. Minimum Property Requirements. The Residence was constructed by the *National Warm Air Heating and Air Conditioning Association* in 1949 to determine if the warm-air perimeter heating system could produce satisfactory results in a residence which did not meet many of the recommended standards of construction. We feel that under the circumstances the warm-air perimeter systems studied have done a remarkable job.

It is impossible to account accurately for the discrepancies between the actual fuel consumption and the calculated heat loss of the Residence. It is desirable, of course, that some reserve furnace capacity be allowed, but it should probably not exceed 10 percent of the input to the furnace. If the reserve of the furnace is too great, it is difficult to maintain uniform room-air temperatures in the residence during mild weather. Part of this discrepancy may be due to the fact that even though the outdoor design temperature in Urbana is −10 F, this temperature is reached only rarely and then only for a short time during the night. Since the fuel consumption

† Building Department, City and County of Denver.

data shown in Table 3 are based on 24 hr average outdoor temperatures, it is inevitable that some solar gain entered into the picture with a consequent reduction in fuel consumption. It is entirely possible that if insulation had been placed in the sidewalls of the Residence, the fuel consumption of the perimeter-loop system would have been closer to that of the perimeter-radial system since the air temperatures in the perimeter duct would have been somewhat lower.

Although the weather conditions experienced during this study have not been defined, the data were obtained during an entire winter, and we had our usual quota of snow, rain, and slush. The site upon which the Residence is located is well drained, and at no time were puddles of water observed standing near the footing wall. Mr. Dill has pointed out that in many locations it would be very desirable that the bottom of the perimeter duct be above the outside grade. I feel that this thought can be carried even further and that in some places it may be desirable that the bottom of the insulation be above the outside grade. If this were done, it would probably be advisable to use an L-type of insulation rather than placing the insulation so that it extended from 18 in. to 2 ft vertically along the foundation wall of the residence.

There is no doubt that this is a type of heating system that requires thorough supervision on the job. Although this statement can be made about any type of heating system, or about any phase of residential construction for that matter, it is especially true for the perimeter-loop and perimeter-radial heating systems. When the warm-air heating ducts are buried in a concrete slab, it is extremely difficult to make alterations and modifications which may be necessary to improve the operation of the heating system. For this reason, the job must be done right the first time.

The use of floor registers has been objected to by many people. For this reason, the warm-air heating industry has developed low-sidewall, baseboard, and extended-baseboard diffusers for use with warm-air perimeter heating systems. It is my personal feeling, however, that the use of floor registers is not as objectionable as Mr. Oslund seems to believe, and that little dust will accumulate during short off-periods of the blower. It might, however, be desirable to clean out the duct system before each heating season. This would remove the summer's accumulation of dust.

The thought behind the design of the perimeter heating system was not so much to reduce the cost of homes in the lower price field by reducing the cost of the heating system as it was to provide a more satisfactory type of heating for the low cost home. It is our feeling that it will usually be more expensive to install ducts in a concrete slab floor than to install an overhead delivery system with the ducts in the attic and the registers either high in the sidewall or in the ceiling. The purpose of perimeter heating, therefore, is not to obtain a heating system at minimum cost, but rather to obtain maximum comfort in a low cost home from a heating system which can be economically installed.

We have conducted no investigations on perimeter heating at the University of Illinois using duct materials other than sheet metal. We do not feel, however, that this is the only material that can be used satisfactorily. Procedures and suggestions for use of other materials are included in Manual No. 4 of the *N.W.A.H.A.C.A.*

The degree of comfort and the stratification in the space expressed in terms of room-air temperature differentials and gradients are functions of the number of outlets installed, their locations, and the temperatures of the floor and other surfaces in the room. This has been shown in University of Illinois Engineering Experiment Station Bulletin No. 401 in which it was stated that more satisfactory room-air temperature conditions were experienced in Research Residence No. 2 when the basement was heated than when it was unheated. In these studies the same register locations were used for both the heated and unheated basement conditions, and the improved room-air temperature conditions were thus directly attributable to the warm floors which were caused by the heated basement. There is no reason to expect that the effect of floor-surface temperatures on room-air temperature conditions should not be the same in a basementless house as it is in a home with a basement.

Although it is true that the isothermal lines in Fig. 7 indicate that there is little air movement in the perimeter loop at the front of the house between the floor registers, I do not believe that the floor-surface isothermal lines should be virtually unchanged if this section of the loop were omitted. Comparing Figs. 7 and 8 of the paper shows that although little air was flowing in that portion of the perimeter loop along the front of the residence, there was considerable difference between the floor-surface temperatures experienced in that region with the loop system and those experienced with the radial system. It should also be pointed out that the loop was in place during the studies of the radial system, but that it was blocked and sealed on both sides of each register.

Even though a system with three or four feeders may be rather common practice in industry for homes the size of Research Residence No. 3, the latest design procedures indicate that if 6-in. diameter feeder and loop ducts are installed it would be necessary to have a minimum of six feeders with a loop system. For an 8-in. diameter system, three feeders would be needed, and five feeders would be needed for a 7-in. diameter duct system. In the case of the earlier studies conducted in Residence No. 3, a three-feeder loop system was used which employed 8-in. diameter feeder and loop ducts. In this earlier study, with the exception of a portion of the kitchen, higher floor-surface temperatures were experienced near the exposed edge of the floor in all sections of the residence than were experienced with the radial system reported in this paper. This work has been reported in University of Illinois Engineering Experiment Station Bulletin No. 403.

It should be quite apparent that the radial system is more readily balanced than is the loop system, but no particular difficulty was experienced in the balancing of either system. When balancing any heating system it is necessary to achieve the desired room-air temperature conditions. When balancing the loop system, care must be taken to achieve the desired room-air temperature conditions by adjusting both the heat entering the rooms through the registers and the heat emission from the warm floors. With the radial system, however, the heat entering the rooms through the registers can be adjusted directly, and changing the damper settings of a register in one room will not affect the room-air temperature conditions in another room.

No attempt has been made in this study to determine the effect of register location on room-air temperature conditions. Furthermore, no attempt has been made to determine the limitations on furniture placement caused by the two types of registers studied. Neither determining the amount of usable space lost when floor registers are employed, nor investigating the possibility of reducing this loss of usable space by employing other types of registers was within the scope of this investigation.

Even though the radial system studied did not conform to the present design procedures, it should be pointed out that these procedures were not published in the 1953 edition of Manual No. 4 of the *National Warm Air Heating and Air Conditioning Association* until after this study had been completed. The most recent design procedures would require the use of not less than eight registers for a radial system installed in Research Residence No. 3. This is based upon a maximum distance between registers of 15 ft. Even if this standard were closely adhered to, it would not be expected that the low floor-surface temperatures which were experienced in the exposed corners of the residence would be entirely eliminated. The two living room registers shown in Fig. 8 are approximately 18 ft apart. It is not felt that decreasing this distance by 3 ft would have a great effect on the floor-surface temperatures in the exposed corners of the living room.

Increasing the number of feeder ducts used with the radial system would probably improve the room-air temperature conditions. The use of additional registers, however, would decrease the register-air velocity enough that the improvement in room-air temperature conditions might be small. It should also be remembered that the use of additional feeder ducts would increase the subfloor heat loss of the residence somewhat and would thus tend to increase the fuel consumption slightly.

1490

No. 1490

HEAT EXCHANGES IN A FLOOR PANEL HEATED ROOM

By L. F. SCHUTRUM*, G. V. PARMELEE**, AND C. M. HUMPHREYS†,
CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THIS PAPER is the second of a series dealing with heat exchanges within panel heated and cooled spaces, studies of which form part of the Society's long-range research program in panel heating and cooling. It describes the results of tests made in the A.S.H.V.E. Environment Laboratory in which the *entire floor area* was the heating panel. Most of the tests were made with all parts of the ceiling and the walls at the same surface temperature (a so-called uniform environment), but some data are given for tests in which these surface temperatures differed from one another, *i.e.*, a non-uniform environment. Results are compared with those of previously reported tests in which the *entire ceiling area* was the heating panel.[1]

TEST APPARATUS

The A.S.H.V.E. Environment Laboratory, equipped and instrumented for direct measurement of heat flow rates through all six surfaces of the room, has been described in a previous paper.[2] The interior surfaces of the room are made up of 75 aluminum panels of various sizes, arranged to provide maximum flexibility in surface temperature control. All room-side surfaces were painted with a semi-gloss gray paint.

Low-inertia, low-resistance heat flow meters, which also have been described previously,[3] cover approximately 10 percent of the inside surface area of the

* Research Fellow, A.S.H.V.E. Research Laboratory. Junior Member of A.S.H.V.E.
** Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.
† Senior Engineer, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

room, the nominal dimensions of which are 24 ft 6 in. x 12 ft with an 8-ft ceiling height.

Surface temperatures were measured by thermocouples cemented in grooves cut into the backs of the aluminum panels.

Air temperatures, measured by butt-soldered, 36-gage, copper-constantan thermocouples, were taken at six locations along two walls and at various levels in the center and near one end of the room.

Infiltration was recognized as an important factor affecting both room air temperatures and panel outputs. The infiltration air, introduced into the room by way of six vertical 2-in. diameter perforated pipes closed at the top, was metered by means of an orifice. Four of the six pipes were located along one long wall (east) and 2 along one short wall (south). Since it was felt desirable to define the manner of exit of this air as completely as possible, the room was fairly tightly sealed and the door cracks were taped. Under such conditions about 50 per cent equivalent of the infiltration air left the room at the floor level through a 6-ft long slot, about 0.4 sq ft in area, in the center of the north half of the west wall. The air temperature was taken at this point. The remaining equivalent of 50 percent of the infiltration air undoubtedly left the room through small cracks between the panels forming the various room surfaces. In a normal house or building there is considerable uncertainty as to the actual entrance and exit temperatures of infiltrating air.

TEST PROCEDURE

All data covered by this report were taken under steady state conditions and mainly with a uniform environment. Panel temperatures and rates and temperatures of infiltration air were controlled; room air temperatures and heat flow rates were the dependent variables.

In this report the term *uniform environment* designates a condition wherein *all panels of the four walls and the ceiling were maintained at the same surface temperature.* The term *non-uniform environment* indicates a condition wherein *ceiling and wall surfaces were not at the same temperature.*

All of the tests described herein were made with the test room unoccupied, unlighted, and containing only the necessary instrumentation.

Calibration: The calibration of the individual heat flow meters and of the Environment Laboratory as a unit was described in a previous paper.[1] The calibrations of selected heat flow meters were checked at frequent intervals during the tests, a further check being maintained by comparing the metered heat inflow from the floor with the metered outflow through the walls and ceiling in those tests in which there was no infiltration. In 19 such tests these heat flows differed by less than 5 percent. In general, the measured heat inflow from the floor was lower than the total outflow through the other surfaces.

TEST RESULTS—UNIFORM ENVIRONMENT

Heat Flow—No Infiltration: Fig. 1 shows the heat output from the upper surface of the floor panel for conditions of uniform environment, no infiltration air, and the room empty and unlighted, as a function of the floor panel temperature and the area weighted average temperature of the unheated surfaces of the room. This *average unheated surface temperature* is abbreviated AUST. The heat transfer coefficient for radiation and convection combined, based upon the

difference between floor temperature and AUST, is about 1.4 Btu per (hr) (sq ft) (F deg) for the range covered in Fig. 1.

A comparison of the heat outputs from this floor panel and from a ceiling panel in the same room yields interesting data. For an AUST of 65 F and a panel surface temperature of 85 F, the floor panel output was approximately 27 Btu per (hr) (sq ft) (see Fig. 1) as compared with a ceiling panel output of 20 Btu (per hr) (sq ft) (see Reference 1). The difference is the increase in convective heat transfer from the floor panel.

Table 1 shows the distribution of heat flow through the cooled surfaces of the room. These data are typical of the tests made without infiltration. It was found that the calculated radiant energy exchanges between the floor and

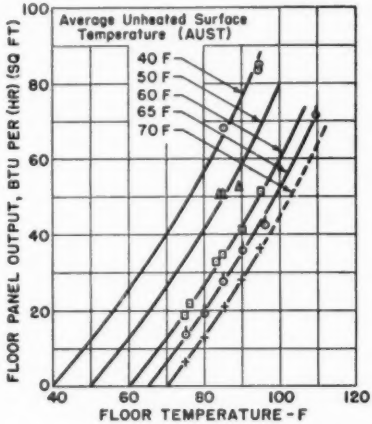


FIG. 1. HEAT FLOW FROM THE FLOOR VS. FLOOR TEMPERATURE AND AUST

(Uniform environment, no infiltration, 24½ x 12 x 8 ft high room)

each of the other room surfaces, expressed as percentages of the sum of these calculated radiant exchanges, were substantially the same as the percentages of total heat exchange shown in Column 4 of Table 1. These percentages are practically the same as those reported for the heated ceiling tests.[1] Note also that the total observed heat flow into the room differed from the observed outflow through the walls and ceiling by 1.2 percent.

The variation in heat transfer through individual room surfaces is illustrated in Fig. 2 which shows the heat flow rate at each heat flow meter location for the south half of the Environment Laboratory. The heat flow rates through the lower part of the walls are greater than those through the upper parts because of the difference in position with respect to the heated panel.

Heat Flow—With Infiltration: Fig. 3 shows the effect of the rate of infiltration on the heat flow through each surface. All surface temperatures and the infiltration air temperature were held constant. It will be seen that as the rate of infiltration increased, the heat output of the panel increased, while the heat

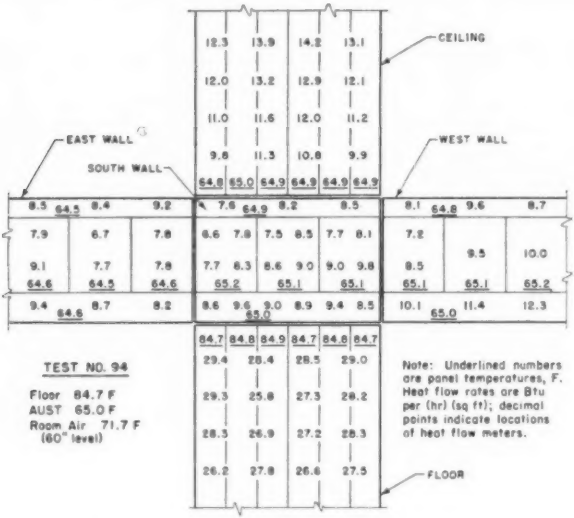

CEILING
EAST WALL
SOUTH WALL
WEST WALL
FLOOR
TEST NO. 94
Floor 84.7 F
AUST 65.0 F
Room Air 71.7 F
(60" level)
Note: Underlined numbers are panel temperatures, F. Heat flow rates are Btu per (hr) (sq ft); decimal points indicate locations of heat flow meters.

FIG. 2. HEAT FLOW AND TEMPERATURE DISTRIBUTION FOR SOUTH HALF OF ENVIRONMENT LABORATORY

(Uniform environment, no infiltration, 24½ x 12 x 8 ft high room)

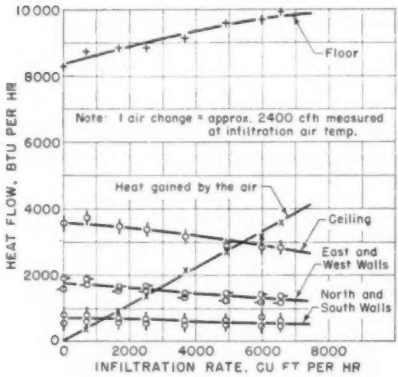

HEAT FLOW, BTU PER HR
INFILTRATION RATE, CU FT PER HR
Floor
Note: 1 air change = approx. 2400 cfh measured at infiltration air temp.
Heat gained by the air
Ceiling
East and West Walls
North and South Walls

FIG. 3. EFFECT OF INFILTRATION RATE ON HEAT FLOW

(Floor at 85 F, all other surfaces at 65 F; 50 F infiltration air temperature; 24½ x 12 x 8 ft high room)

outflow through the other five (cooled) surfaces decreased. These changes are the result of the combined effect of a decrease in room air temperature and an increase in convection conductances.

Table 2 gives the heat flow through each surface of the room for one of the tests shown in Fig. 3. Surface temperatures were essentially the same as those in Table 1, for a test without infiltration.

Fig. 4 shows the additional heat output from the floor due to infiltration air in terms of the infiltration air rate and temperature and of the AUST. These values, added to the appropriate values from Fig. 1, give the total output of the floor panel when there is infiltration into the room.

Heat Transfer to the Air: With no infiltration, the room air temperature is established by a heat balance between the convective heat gain from the surfaces warmer than the air and the convective heat losses to the colder surfaces. Radiation absorption by gases and water vapor in the air also enters into this balance. Nevertheless, the net heat gain by the room air is zero under steady state conditions. With infiltration, however, some heat is required to warm the incoming air and this heat is removed from the room by the exfiltrating air. This heat quantity is termed *heat gain by infiltration air* for the remainder of this paper.

The heat gain by the infiltration air is found from the following energy balance:

$$\text{Heat input from floor} = \text{Net heat outflow through other surfaces} + \text{Heat gain by the infiltration air} \quad \ldots\ldots\ldots \quad (1)$$

The satisfactory heat balances obtained in the tests with no infiltration indicate that the heat gain by the air could be obtained from the above equation with adequate accuracy. The data given in Table 2 were determined in this way.

The temperature of all the air leaving the room could not be measured directly. Approximately half of it left through the slot where temperature was measured.

Calculations based upon the heat gain by infiltration air as determined by the method described above showed that the average temperature of the exfiltration air was practically equal to the average of the exit air temperature measured at the slot and the area weighted average temperature of all room surfaces

TABLE 1—HEAT FLOW DISTRIBUTION FOR UNIFORM ENVIRONMENT WITH NO INFILTRATION

Test No. 94 (AUST: 65.0 F); Air Temperature at Center of Room: 71.8 F at 30-in. level; 71.5 F at 60-in. level

COL 1 SURFACE	COL 2 SURFACE TEMP F	COL 3 TOTAL HEAT FLOW Btu/hr	COL 4 PERCENTAGE OF FLOOR HEAT FLOW	COL 5 HEAT FLOW RATE Btu/(hr) (sq ft)
Floor	84.7	8385	100.0	27.9
Ceiling	64.9	3615	43.1	12.0
North Wall (incl door)	66.4	570	6.8	6.0
South Wall	64.7	785	9.4	8.2
East Wall	64.7	1605	19.1	8.2
West Wall	64.8	1915	22.8	9.8
Total (Walls and Ceiling)		8490	101.2	

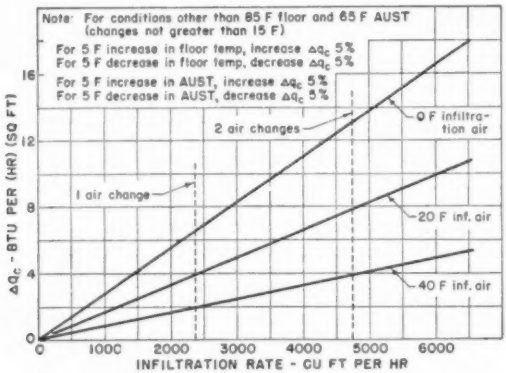

Note: For conditions other than 85 F floor and 65 F AUST (changes not greater than 15 F)
For 5 F increase in floor temp, increase ΔQc 5%
For 5 F decrease in floor temp, decrease ΔQc 5%
For 5 F increase in AUST, increase ΔQc 5%
For 5 F decrease in AUST, decrease ΔQc 5%
ΔQc - BTU PER (HR) (SQ FT)
16
12
8
4
0
2 air changes
1 air change
0 F infiltration air
20 F inf. air
40 F inf. air
0
1000
2000
3000
4000
5000
6000
INFILTRATION RATE - CU FT PER HR

FIG. 4. ADDITIONAL HEAT OUTPUT FROM FLOOR DUE TO INFILTRATION AIR

(Floor 85 F, all other surfaces 65 F; 24½ x 12 x 8 ft high room)

(hereafter referred to as the AST). For these tests, then, the heat gain by the air can be related to the controlled variables: room surface temperature, infiltration air rate and temperature. This relationship is shown in Fig. 5.

The Effect of Point of Entrance and Exit of Infiltration Air: As previously described, the infiltration air normally entered the room at four points along one long wall (east) and at two points along one short wall (south). About 50 percent of this air left the room by way of an exit slot in the floor adjacent to and centered along the north half of the west wall. In a series of tests in which surface temperatures and infiltration air temperatures were held constant, the points of air supply were varied, the infiltration rate being maintained at 1½ air changes. First, only the two supply points along the south wall were

TABLE 2—HEAT FLOW DISTRIBUTION FOR UNIFORM ENVIRONMENT WITH INFILTRATION OF TWO AIR CHANGES AT 39.2 F

Test No. 130 (AUST: 65.1 F); Air Temperature at Center of Room: 68.4 F at 30-in. level; 68.3 F at 60-in. level; Air Temperature at Exit Slot: 69.2 F

COL 1 SURFACE	COL 2 SURFACE TEMP F	COL 3 TOTAL HEAT FLOW Btu/hr	COL 4 PERCENTAGE OF FLOOR HEAT FLOW	COL 5 HEAT FLOW RATE Btu/(hr) (sq ft)
Floor	84.7	9605	100.0	31.9
Ceiling	65.2	2870	29.9	9.5
North Wall (incl door)	66.2	500	5.2	5.2
South Wall	64.8	625	6.5	6.5
East Wall	65.0	1265	13.2	6.5
West Wall	64.7	1460	15.2	7.4
Heat Gain by Air by difference		2885	30.0	

used, thus giving a long flow path to the exit slot. The flow path was then shortened by using only the two supply points centered on the north half of the east wall.

These two variations of the infiltration points of entrance had no significant effect upon the heat flow from the floor panel nor on the room air temperature at the 30-in. and 60-in. levels (center of room). The heat gained by the infiltration air, however, was slightly higher than normal when the long air-flow path was used and slightly lower when the short path was used.

In other tests the room was tightly sealed so that all the air left by means of the exit slot. This was checked by actual measurement. In these tests all six infiltration entrance points were used. The heat flow from the floor panel,

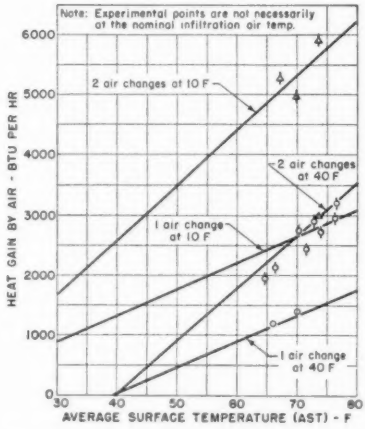


FIG. 5. HEAT GAIN BY INFILTRATION AIR

(Uniform environment, 24½ x 12 x 8 ft high room)

the room air temperature, and the heat gained by the infiltration air showed that no significant changes resulted from sealing the room surfaces against air leakage. With this arrangement a complete energy balance could be made for the tests with infiltration as well as for those without infiltration.

Room Air Temperature—No Infiltration: The room air temperature gradient from the floor to the ceiling at the center of the room is shown in Fig. 6. The dimensionless abscissa scale generalizes the use of this curve, which shows the average values for 23 tests. With this dimensionless coordinate the room air temperature at any level can be found from the equation:

$$\text{Room air temperature} = (\text{A constant})\,(\text{floor temp minus AUST}) + \text{AUST} \quad . \; . \; . \quad (2)$$

The constant varies with height in accordance with the curve. For example, the constant is 0.32 for the 60-in. level. The maximum deviation of the experimental data from the curve was about 5 percent for the 12-in. and higher levels and about 10 percent for levels below 12 in. The second abscissa scale shows

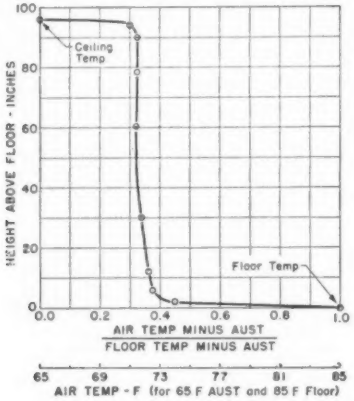


FIG. 6. FLOOR TO CEILING TEMPERATURE DISTRIBUTION AT THE CENTER OF ROOM

(Heated floor, uniform environment, no infiltration, 24½ x 12 x 8 ft high room)

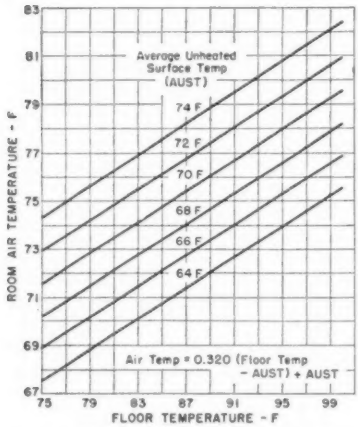


FIG. 7. AIR TEMPERATURE AT 60-IN. LEVEL, CENTER OF ROOM, VS. FLOOR TEMPERATURE AND AUST

(Uniform environment, no infiltration, 24½ x 12 x 8 ft high room)

the variation in room air temperature with height for the specific conditions of 65 F AUST and 85 F floor temperature.

Fig. 7, which shows the room air temperature at the 60-in. level *vs.* floor temperature and AUST, was plotted by means of Equation 2.

Room Air Temperature—With Infiltration: Infiltration caused the room air temperature at the 60-in. level (center of room) to decrease in the manner shown in Fig. 8 for the given conditions. As expected, the room air temperature decreased as the infiltration air temperature decreased and the infiltration rate increased.

The room air temperature gradients for tests with infiltration are compared in Fig. 9 with the average no-infiltration curve of Fig. 6. The infiltration air-flow rate and temperature were the only controlled variables which were

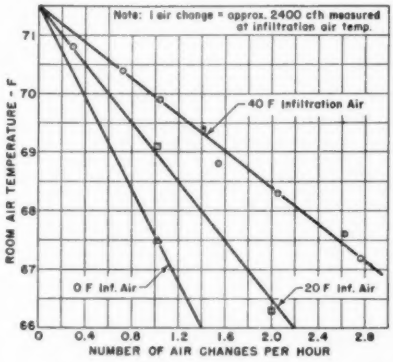


FIG. 8. THE EFFECT OF INFILTRATION AIR RATE AND TEMPERATURE ON ROOM AIR TEMPERATURE

(Room air temperature taken at 60-in. level, center of room; floor 85 F; all other surfaces 65 F; 24½ x 12 x 8 ft high room)

changed in this series of tests. Test No. 130 (Fig. 9) showed a marked reduction in air temperature when two air changes at 39.2 F were introduced. A still lower room air temperature resulted when the infiltration air temperature was reduced to 21.6 F as shown by test No. 129.

Fig. 10, based on data from 19 tests, shows the reduction in room air temperature at the 60-in. level (center of room) due to infiltration air. The greatest deviation between experimentally determined values and those determined by combining the curves of Fig. 7 and the values of Fig. 10 was 1.1 F deg.

The Effect of Average Wall and Ceiling Conductance: Let the conductance C' be defined as a total heat flow through a wall divided by the product of the area of the wall and the temperature difference between the indoor surface of the wall and the outdoor air. Then the cooled surface of the test room can represent an actual exposed wall whose average conductance, for a given heat flow rate, depends on the value assigned to the difference between the temperatures of the indoor surface and the outdoor air. For example, a heat flow rate of 10 Btu per

(hr) (sq ft) and a surface temperature of 65 F could represent combinations of conductance and outside air temperature respectively of 0.5 Btu per (hr) (sq ft) (F deg) and 45.0 F, 0.20 and 15 F, or any combination in which the product of conductance and temperature difference equalled 10 Btu per (hr) (sq ft).

The effect of an average conductance on the AUST and on the floor temperature and heat flow required to maintain an air temperature of 70 F at the 60-in. level (center of room) is shown in Fig. 11 for conditions of 0 F, outside air and one air change, based upon the air density at 0 F. Note that for values of C' greater than about 0.05 the AUST is below 70 F and heat flow is from the air to the walls. For values of C' less than 0.05 the opposite is true. *It is*

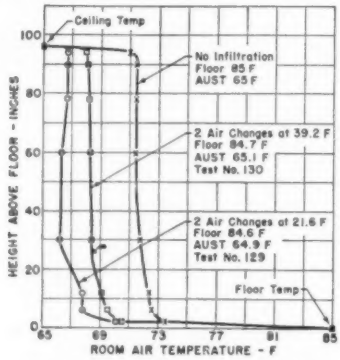


FIG. 9. THE EFFECT OF INFILTRATION ON FLOOR TO CEILING TEMPERATURE DISTRIBUTION AT THE CENTER OF THE ROOM

(Uniform environment, 24½ x 12 x 8 ft high room)

important to realize that Fig. 11 applies only to the room and to the conditions noted.

TEST RESULTS—NON-UNIFORM ENVIRONMENT

Tests were made with various combinations of ceiling and wall temperatures to explore the performance of the floor panel with a non-uniform temperature environment. The results are shown in Table 3. Columns 2 to 7 in the table describe the non-uniform experimental conditions. In columns 8 and 9 the panel heat output for the particular test is compared with the value which would have been obtained in the room with the same AUST and floor temperature but with a uniform environment (see Figs. 1 and 4). In all the tests the panel output for a non-uniform environment was greater than that for the uniform environment.

The last four columns of Table 3 compare measured room-air temperatures for the non-uniform environment tests with calculated temperatures for a uniform environment for the same AUST, panel temperature and infiltration conditions

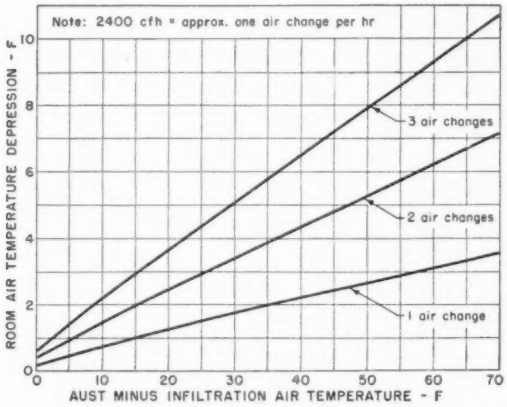


FIG. 10. DECREASE IN ROOM AIR TEMPERATURE AT THE 60-IN. LEVEL, CENTER OF ROOM, WITH INFILTRATION

(Uniform environment, 24½ x 12 x 8 ft high room)

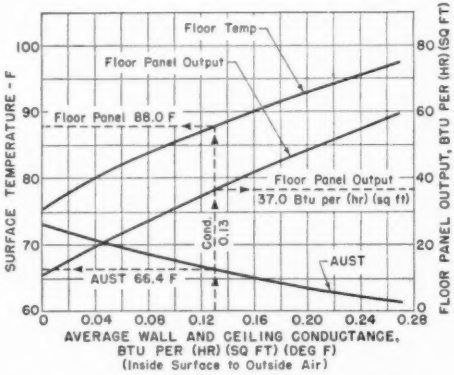


FIG. 11. THE EFFECT OF AVERAGE WALL AND CEILING CONDUCTANCE ON THE AUST AND ON THE FLOOR TEMPERATURE AND HEAT OUTPUT REQUIRED TO MAINTAIN 70 F ROOM AIR TEMPERATURE AT THE 60-IN. LEVEL

(0 F outside temperature, one air change at 0 F, 24½ x 12 x 8 ft high room)

(see Figs. 7 and 10). The data show that the non-uniform environments had little effect on room air temperature.

The heat gain by the infiltration air for the non-uniform environment tests with infiltration (Nos. 172 and 175) averaged about 10 percent higher than the gain for the comparable uniform temperature conditions.

CONCLUSIONS

It is recognized that most of the experimental work described in this paper was carried out under conditions seldom, if ever, encountered in normal practice, *i.e.*, the room unfurnished and unoccupied, all parts of the ceiling and walls at the same surface temperature, and the entire floor area heated to a uniform temperature. However, tests of this type are considered essential for the development of basic information, to be used with data from tests which are now being made under conditions more nearly representative of practical operating conditions for floor and ceiling-panel heated rooms.

It should be remembered that the method given in the HEATING VENTILATING AIR CONDITIONING GUIDE for determining the required panel temperature for a floor or ceiling panel uses a procedure in which the average unheated surface temperature (AUST) of the room is calculated and the temperature of the panel determined from charts in which the AUST is one parameter and the required panel output the other. In other words, the equivalent of a uniform environment is established by calculation, and panel performance is related thereto. To this extent the data developed in these studies and presented here are applicable for design purposes.

However, other variables may influence panel output and room air temperature. In the tests under way as this paper is written, the effects of furnishings,

TABLE 3—THE EFFECT OF NON-UNIFORM ENVIRONMENT ON FLOOR HEAT OUTPUT AND ON ROOM AIR TEMPERATURE

Test No.	Surface Temperature, F				Infiltration Air		Floor Heat Output Btu/(hr) (sq ft)		Room Air Temperature Center of Room, F			
	Floor	Glass (East Wall)	All Other Surfaces	AUST	Changes Per hr	Temp F	Non-Uniform Conditions Test Values	Uniform Conditions Calculated Values	Non-Uniform Conditions Test Values		Uniform Conditions Calculated Values	
									30-in.	60-in.	30-in.	60-in.
Col 1	Col 2	Col 3	Col 4	Col 5	Col 6	Col 7	Col 8	Col 9	Col 10	Col 11	Col 12	Col 13
164	84.6	45.3	65.2	60.8	None		35.2	32.8	68.4	68.4	68.8	68.4
167	85.3	36.2	64.5	58.1			39.3	37.2	66.7	66.7	67.3	66.8
171	84.5	36.0	69.4	62.5			34.2	30.5	69.4	69.6	70.0	69.6
170	84.8	35.8	74.4	65.9			29.7	26.5	72.5	72.8	72.3	72.0
168	84.6	26.0	65.0	56.4			43.6	39.5	65.2	65.2	65.9	65.4
165	75.1	45.9	65.0	60.7			20.6	18.8	65.3	65.3	65.6	65.3
166	74.8	36.3	64.4	58.3			24.4	22.2	63.5	63.7	63.9	63.6
169	75.3	26.4	65.3	56.8			29.9	24.5	62.2	62.3	63.0	62.7
175	84.7	45.6	70.0	64.7	2	40	34.1	31.6	66.9	67.9	68.2	68.2
174	85.2	45.6	70.4	64.8	1	40	32.9	29.5	70.0	70.0	70.1	69.8
172	84.7	36.1	69.2	62.0	1	40	37.4	32.9	68.1	68.1	68.2	67.9
173	84.9	36.2	70.1	62.8	1	zero	40.6	36.8	64.8	66.7	66.8	66.7

lights, and floor coverings are being studied, not only under conditions of uniform environment, but also under various non-uniform conditions, one of which, for example, incorporates two simulated outside walls in which there are three simulated windows.

While limitations, imposed by the conditions under which the tests were made, are recognized, the following observations are justified:

1. Pending the establishment of data on the radiative and convective components of the total heat output from floor panels, studies of which are currently under way, the values for total panel output obtained in these tests indicate that the values given in THE GUIDE 1952 (Fig. 11, p. 548) are of the right order of magnitude for unheated mean radiant temperature (UMRT) values of around 65 F.

2. Room air temperatures were found to be higher for a floor panel heated room than for a ceiling panel heated room for the same room surface temperatures. For example, with an 85 F panel temperature and a 70 F AUST, with no infiltration, the room air temperature at the 60-in. level would be 74.8 F with a floor panel, and 71.2 F with a ceiling panel.

3. The effect on the room air temperature at the 60-in. level of varying the amount of infiltration air and its entering temperature was about the same in the floor panel tests (see Fig. 10 this paper) as it was shown to be in the ceiling panel tests (see Fig. 9, Reference 1).

4. In none of the tests reported on here did the air temperature gradient in the room between the 2-in. and the 90-in. level exceed 3.5 F deg.

REFERENCES

1. RESEARCH REPORT No. 1473—Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 197).

2. RESEARCH REPORT No. 1444—The A.S.H.V.E. Environment Laboratory, by Cyril Tasker, C. M. Humphreys, G. V. Parmelee, and L. F. Schutrum (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 139).

3. RESEARCH REPORT No. 1453—A Low-Inertia Low-Resistance Heat Flow Meter, by R. G. Huebscher, L. F. Schutrum, and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 275).

DISCUSSION

P. B. GORDON, New York, N. Y.: I want to thank the authors and particularly Mr. Schutrum for a good paper and a fine presentation. This is a companion paper to one presented in Chicago last January on ceiling panel heat output and is similar in method and format. There are several subsequent papers which are being prepared describing studies now underway and contemplated.

Such studies include an investigation of non-uniformity. In the present paper conditions of uniform environment were maintained although the effects of non-uniformity were mentioned. These studies on non-uniform environment are underway for both ceiling and floor panels. A second group of studies will consider the effects of panel performance when room size and shape are changed. A third problem to be considered will be the effect of room furnishings and floor coverings on output for ceiling and floor panels.

Another area to be investigated is separating the radiation component from the convection component in the total output values. In this paper these outputs are given

as total values and data that will separate the radiation exchange from the convection exchange at the surface are important.

It might be of some further interest to note that the laboratory has been engaged in revising piping connections to permit similar studies on cooling by means of panels.

The values of Fig. 1 in this paper are not for a 70 F room, but for a room wherein the air temperature results from the particular experiment underway. Refer to Fig. 7, and assume that most rooms in normal practice would operate at AUST between 60 and 70 F and usually in the order of 65 F plus or minus 3 deg. In such a case the air temperature in these experiments would be 70 to 71 F plus or minus 2 deg.

It should not be assumed from the foregoing that air temperature cannot be raised to 70 F if designed for such a temperature. In these particular experiments with both independent and dependent variables, the resultant air temperature is a dependent value. By calculating the required Btu input the air temperature can be raised to any level desired.

Using the data given in Fig. 1 (assuming an 85 F floor temperature), with an AUST of 65 F, the output obtained is 27 Btu per (hr) (sq ft). With a 60 F AUST the output is 32 Btu per (hr) (sq ft).

In Fig. 4 data are given for infiltration effect which shows an increase in heat output. For a 65 F AUST, and one half air change per hour, the increased output is about 4 Btu per (hr) (sq ft); assuming one air change per hour the increased output equals 6 or 7 Btu per hr.

The data given in Table 3, columns 8 and 9, show that in an experimental run with average room conditions the difference between uniform conditions and non-uniform conditions is about 2 to 4 Btu per (hr) (sq ft). Assuming 3 Btu as the average, with an AUST of 65 F, there is an average total output of 34 Btu per (hr) (sq ft) (*i.e.*, 27 plus 4 for infiltration plus 3 for non-uniformity), and for an AUST of 69 F an output of 41 Btu per (hr) (sq ft) is obtained.

Thus, projecting the findings, and considering all the changes resulting from infiltration and non-uniformity we can obtain data for heat outputs in the laboratory for floor panel systems. Similar work, of course, will be done for ceiling panel systems.

The data of this paper, supplemented by the investigations now under way, will enable us to obtain a very exact expression to determine the values for outputs under many different conditions. Such information will explain many of the discrepancies that have always concerned designers and builders of panel systems.

A. B. ALGREN, Minneapolis, Minn.: I would like to preface my comments with two statements; first, the authors are to be highly commended in presenting basic information that should provide some of the answers to the controversial subject, Floor Panel Heating; secondly, that our membership is to be commended for their part in supporting the furtherance of our research activities and facilities. Our environment room is an excellent example, for this facility offers the means for providing both basic studies and a simulated field application.

As indicated in this paper one important factor in panel output is the effect of convection. Although we have been conducting field studies on panel heating systems for several years, the panel output values that were determined represent the panel output for that particular installation. It has been common practice to base the convection output entirely on air temperature. From this investigation it is evident that in addition to the air temperature, an important factor to be considered is the effect of a uniform or non-uniform environment. Although the AUST is the same, the convection output would be greater for a non-uniform environment. This would indicate that further studies should be made with non-uniform environment and vary the ratio of area maintaining the same AUST.

Another important factor would be the location of these variable temperature areas, in the vertical or horizontal plane. Recently, in one of my field studies, I had occasion to investigate the effect of varying the thermal resistance of the ceiling in relation

to the thermal resistance of the wall. The results indicated a decrease in panel output, with improved comfort conditions.

Although THE GUIDE values based on calculated values supplemented by field test data represent reasonable design values in the range for which they are commonly used, the results of this investigation indicate the need for further studies on radiative and convection components. The authors have stated that these studies are now in progress. When they are completed, and reported, I believe that the answers to the many factors on floor panel output will be answered.

W. A. DANIELSON, Raleigh, Tenn.: Recently a one-story office building was constructed in Memphis. It was well built and had a 6-in. concrete slab floor on top of gravel. On a cold day it requires considerable time to heat the building because of the mass of the floor. Such heating causes the clerks to complain about hot feet. If the weather has moderated on the following day, it is necessary to open the doors and windows to admit outside air to get the office comfortable. The removal of the heat stored in the heavy floor slab requires many hours.

C. G. NEWTON, Chicago, Ill.: I would like to add my congratulations to the authors of this paper. There is a great deal of material in a report of this kind which has taken months of preparation, and it requires more than a few minutes of study to be able to comment on it comprehensively. There are, however, a few suggestions which might be made on this paper and on the possible future studies which are to follow.

The uncomfortably high floor temperatures which have been discussed here, together with the suggestion that floor temperatures should be arbitrarily limited, might indicate that a very fruitful source of study could be on the physiological aspects of the results of occupancy of a panel heated environment. We had a very interesting paper the other day dealing with physiological reactions to environment. It might be developed as a result of even a preliminary study along those lines that the Society should establish fairly rigid limits to panel surface temperatures, with the understanding that when total heat losses for the installation cannot be met without exceeding these limits, some supplementary heating means must be provided.

Before too many final conclusions have been drawn as to the actual total heat release from a panel, perhaps more time should be given to the study of the nonuniform environment. It is entirely probable that in such a study many of the variables which have been mentioned in this paper can be tracked down. Our experience in the field, particularly with the larger installations, would indicate that we are overdesigning our systems no matter how carefully and conscientiously we attempt to avoid such overdesign. The result is more serious in a panel heating system than in the more conventional systems, because the panel system that is out of balance, or is not sensitive to temperature changes, introduces very serious problems of control. I do not want to imply that we should be careless in our application of the knowledge that we have already gained regarding panel heating systems, but I would suggest that we should be very cautious about adding heavy factors of safety to our calculations. If anything, we should lean in the other direction.

These studies deal primarily with a single aspect of a heating system. They are very carefully and completely developed, but little has been done in the study of the over-all performance of a system in which panels, or radiators or other heating devices are to be components. As a consequence, it seems that we are inclined many times to overdesign because of lack of consideration for all of the factors that will effect the performance of the system. A study of this kind will, of course, introduce many new variables, and lead to additional questions, but until we are able to actually evaluate one system and one method of producing heat in or controlling a panel, against the results to be obtained in other systems, we are probably going to find that our design calculations are not entirely accurate in the light of eventual experience.

AUTHORS' CLOSURE (L. F. Schutrum): Environment Laboratory experiments involving temperature non-uniformity of unheated surfaces, room furnishings, and overall carpeting have been completed, and analysis of the data will be reported soon.

As Professor Algren mentioned, common practice is to base convection output on room air temperature. Convective heat transfer for conditions set forth in this paper can be determined by subtracting calculated radiative heat transfer from overall heat transfer as in Fig. 1. The convection coefficient of heat transfer can, in turn, be based on a room air temperature determined from information given in the paper.

Separation of total heat transfer into radiation and convective components awaits the determination of the emissivity of the room surfaces. Sometime ago, the equipment for this investigation of surface emissivity was set up, and the procedure and techniques developed so that both normal and angular emissivities up to about 85 F can be measured.

An investigation has started on the effect of room size on panel performance. The room size was 12 ft × 24½ ft with an 8 ft ceiling for the ceiling panel heated room, previously reported, and this floor panel heated room. Recently a partition was installed in the center of the Environment Laboratory dividing it into two 12 ft × 12 ft × 8 ft high rooms. To complete this series of tests on the effect of room size on panel performance, the Environment Laboratory will be extended to its full size, 12 ft × 24½ ft with a 12 ft ceiling.

The import of the panel heating control problem mentioned by General Danielson was recognized when the Society initiated the overall panel heating and cooling research program. Experiments are now being conducted on this matter under the guidance of Group D of the Technical Advisory Committee on Panel Heating and Cooling. The results of these studies may supply some of the answers to Mr. Newton's suggestions.

The Technical Advisory Committee on Sensations of Comfort has been charged with the problem of developing data on comfort conditions in panel heated and cooled spaces. This committee is waiting for the termination of the present studies conducted in the Environment Laboratory so that it, too, can use these facilities.

1491

No. 1491

FURTHER STUDIES OF THE THERMAL CHARACTERISTICS OF PLASTER PANELS

By L. F. SCHUTRUM* AND C. M. HUMPHREYS**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

WHEN THE FIRST results of studies at the A.S.H.V.E. Research Laboratory on the thermal characteristics of plaster panels were published[1] in July 1951, it was questioned whether the four panels tested were truly representative of plaster panels found in practice, particularly with regard to tube embedment. The chief reason for doubt seemed to be that the heat output of the panels was somewhat at variance with those normally used to design ceiling panel heating systems. The first four panels which had been built to represent three conventional types, included one panel with tubes below gypsum lath, one with tubes below expanded metal lath, and two with tubes above expanded metal lath.

In an attempt to settle the question raised, it was decided to build two additional panels, with tubes placed above metal lath, one of which would have unusually good tube embedment, the other rather poor tube embedment.

This report covers the results obtained with these two panels and compares performance with that obtained with the other four panels. All these studies were made under the advisory guidance of Group A△ of the Technical Advisory

* Research Fellow, A.S.H.V.E. Research Laboratory. Junior Member of A.S.H.V.E.

** Senior Engineer, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

△ *Personnel:* R. S. Dill, *Chairman;* H. L. Flodin, F. E. Giesecke, G. D. Lain, R. L. Maher, V. L. Miller, D. L. Mills, C. W. Nessell, F. E. Parsons, S. I. Rottmayer, R. T. Schoerner, E. E. Scott, S. K. Smith, R. K. Thulman, S. M. Van Kirk, L. H. Yeager.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Denver, Colo., June 29-30, July 1, 1953.

Committee on Panel Heating and Cooling as a part of the Society's overall research program, organized to develop design data for panel heating and cooling systems.

PANEL CONSTRUCTION

Two panels were constructed on wood frames 6 ft square, with joists in the test section located on 16-in. centers. Each panel contained seventeen ⅜-in. nominal (½-in. O.D.) nonferrous tubes, located on 4-in. centers above the

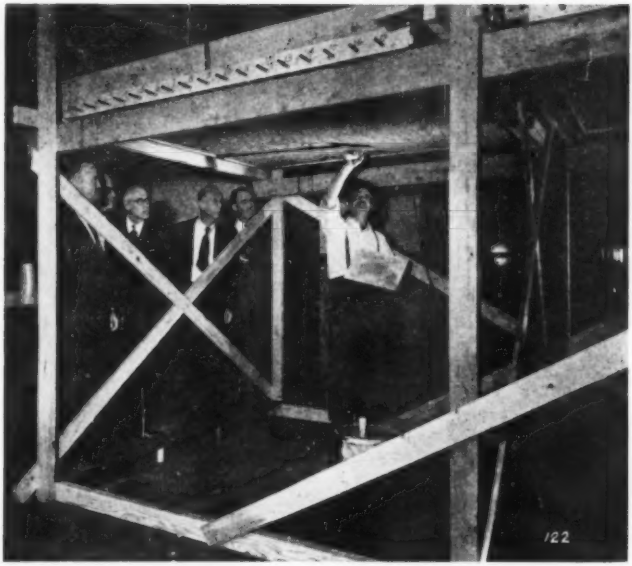


FIG. 1. APPLICATION OF PLASTER TO TEST PANEL

metal lath. *The lath was wired securely to each tube at 8-in. intervals.* The plaster was applied in three coats: scratch, brown, and finish, to a depth of ¾-in. below the lath. The first (scratch) coat, which consisted of one part unfibered neat gypsum and two parts sand, by weight, penetrated the lath and partially embedded the tubes. The second (brown) coat consisted of one part unfibered neat gypsum and three parts of sand, by weight. The finish coat consisted of three parts lime putty and one part gaging plaster, by volume. These plaster mixes conformed to the standard specification for gypsum plastering.[2] Plastering was done by an experienced craftsman with the panel frames supported overhead in a horizontal position (Fig. 1). The plastering operation was observed by a number of Committee members who expressed the view that one panel (No. 5), had much better embedment than found in practice, and that the other (No. 6), was nearly as good as that generally found in the field.

Panels Nos. 5 and 6 differed in one important respect from panels of similar construction, Nos. 1 and 3, used in earlier studies.[1] In panels Nos. 5 and 6, the lath and tubes were wired together at 8-in. intervals along each tube; similar ties had been inadvertently omitted from earlier panels. Another variation was in the type of lath used. Panel No. 1 had flat rib metal lath weighing 2.5 lb per sq yd; panels Nos. 3, 5, and 6 had diamond key type metal lath weighing about 3.1 lb per sq yd. It is not believed that the difference between the two types of metal lath was of any significance in effect on the results.

TEST APPARATUS AND PROCEDURE

Test apparatus, instrumentation, and procedures were the same as those used in the previous studies.[1] The panel to be tested was placed between two heat absorbers, as shown in Fig. 2. It was possible to control the heat flow rate from each side of the panel by maintaining the heat absorber temperatures at

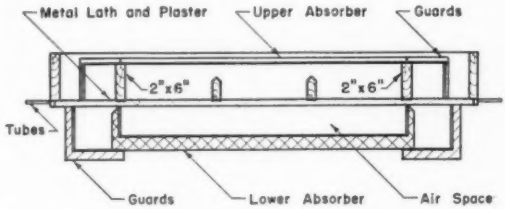


FIG. 2. SECTIONAL VIEW OF PLASTER PANEL AND ABSORBERS

the proper levels. Heat flow rates were measured by heat flow meters incorporated in the absorber construction. Guards were placed along the four sides of each absorber to eliminate edge losses. As an additional check on the heat flow measurements, two plate-type heat flow meters[3] (each 11½ in. sq) were installed in each of the two absorbers. The metering accuracy of the two absorbers, determined by electrical calibration, was found to be within 5 percent of the measured input.

The tubes or pipes embedded in each panel were connected to headers to form a grid-type coil. By selective valving of key cocks provided on each tube, tests were made with 4, 8, 12, and 16-in. tube spacings. The temperature of the water supplied to the panel could be held constant at any desired value. In general, each series of tests was made with tube temperature held constant, and with absorber temperatures adjusted to produce heat flow rates of the desired magnitude. Temperature distribution and heat flow rates from both sides of the panel were determined for each test.

Tests were made without insulation on the back of the panel and also with 3⅝-in. batt-type insulation, with a nominal conductivity of about 0.27 Btu per (hr) (sq ft) (F deg per in.), laid on the back of the plaster.

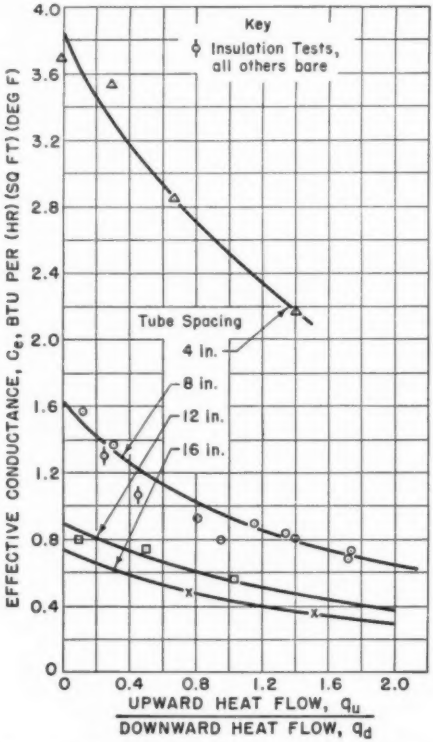


FIG. 3. EXPERIMENTAL VALUES OF EFFECTIVE CONDUCTANCE FOR PLASTER PANEL NO. 5

EFFECTIVE PANEL CONDUCTANCE

Following the pattern used previously,[1] the performance of each of panels Nos. 5 and 6 is reported in terms of an effective panel conductance. This is defined as the downward heat flow in Btu per (hr) (sq ft) divided by the difference between the average tube temperature and the average panel surface temperature (F deg), and is a function of the tube spacing and of the ratio of upward to downward heat flow. Performance curves for plaster panels Nos. 5 and 6 are given in Figs. 3 and 4 which show that the panel with good tube embedment, No. 5, has a higher effective conductance than the one with poor tube embedment, No. 6. This comparison may also be seen in Fig. 5 and Table 1. Insulation placed on the back of the plaster did not appreciably change the effective conductance for a fixed ratio of upward to downward heat flow. (See tests with 8-in. tube spacing—Figs. 3 and 4).

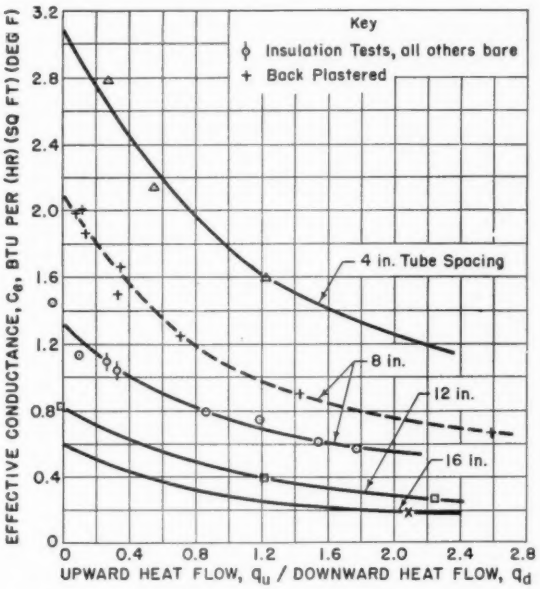


FIG. 4. EXPERIMENTAL VALUES OF EFFECTIVE CONDUCTANCE FOR PLASTER PANEL NO. 6

EFFECT OF BACK-PLASTERING

After completion of the initial work on panel No. 6 having poor tube embedment, plaster was applied uniformly to the back of the panel to a depth of $\frac{3}{8}$ in. over the tubes. The plaster used was similar to that of the brown (second) coat but included fibrous material. When the plaster was properly cured, tests were made to determine the effect of back-plastering on panel performance.

The experimental effective conductance of panel No. 6 when back-plastered is given in Fig. 4 for an 8-in. tube spacing. This effective conductance is approximately equivalent to that of the same panel with a 6-in. tube spacing without back-plastering.

In the original studies, back-plastering was applied to a depth of about 3/16 in. over the tubes and tapered down to the metal lath. The effect of this back-plastering upon the effective conductance of the panel was about the same as reported here for panel No. 6.

SURFACE TEMPERATURE

The surface temperature along a line perpendicular to the tubes has a regular wave form showing highest temperatures directly below the tubes and lowest temperatures midway between tubes. The surface wave amplitude (half the

peak to peak value) depends upon the total heat flow from the panel and is given for various tube spacings in Fig. 6 for panel No. 5, and in Fig. 7 for panel No. 6. Panel No. 5 has the smaller surface wave amplitude for all tube spacings.

PERFORMANCE COMPARISON OF SIX PLASTER PANELS

Table 1 gives pertinent data on the construction and the thermal characteristics of all six panels and lists the panels in order of decreasing effective conductance. Thermal characteristics may be compared on the basis of effective

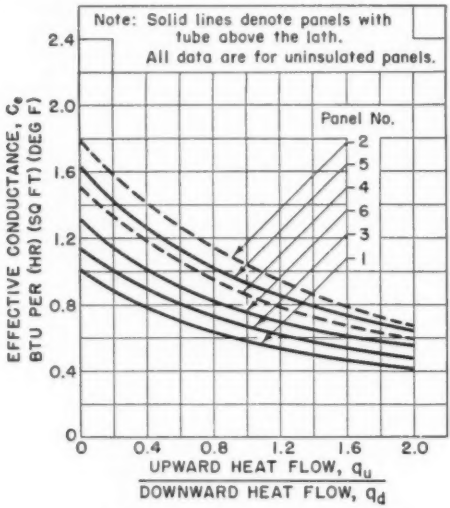


FIG. 5. EFFECTIVE CONDUCTANCE FOR SIX PLASTER PANELS: 8-IN. TUBE OR PIPE SPACING

conductance; this is shown in Fig. 5 based on an 8-in. tube spacing for uninsulated panels. Table 1 and Fig. 5 show that panel No. 6, with relatively poor tube embedment, has a higher conductance than panel No. 1 which had at least average tube embedment. These results emphasize strongly the importance of secure fastening of the tubes to the lath.

UPWARD HEAT FLOW

In any application, the downward heat flow and the panel surface temperature requirements are determined by the heating requirements of the space. However, the effective conductance, and therefore the tube temperature, depends upon the ratio of upward to downward heat flow; consequently, it is necessary to know both the upward and the downward heat flow to determine the effective conductance of the panel.

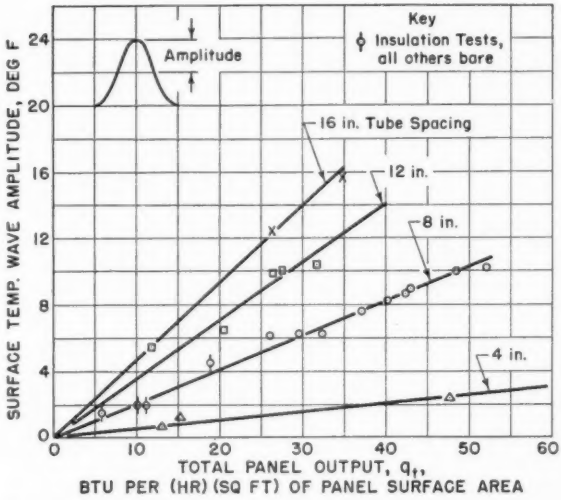


FIG. 6. SURFACE TEMPERATURE WAVE AMPLITUDE VS. PANEL OUTPUT: PANEL NO. 5

TABLE 1—EFFECTIVE CONDUCTANCE OF PLASTER PANELS

(Listed in order of decreasing effective conductance)

PANEL NO.	SIZE AND TYPE[a] OF TUBING INCHES	LOCATION OF TUBES	TYPE OF LATH	DEPTH[d] OF COVER INCHES	TUBES WIRED TO LATH	TUBE EMBEDMENT	EFFECTIVE CONDUCTANCE[e] C_e Btu/(hr) (sq ft) (F DEG)
2	⅜ nonferrous	Below lath	Expanded metal[c]	⅜	—	Almost complete	1.58
5	⅜ nonferrous	Above lath	Expanded metal[b]	¾	8-inch intervals	Good	1.42
4	⅜ nonferrous	Below lath	Gypsum lath	⅜	—	Almost complete	1.33
6	⅜ nonferrous	Above lath	Expanded metal[b]	¾	8-inch intervals	Poor	1.14
3	½ ferrous pipe	Above lath	Expanded metal[b]	¾	none	Average	1.00
1	⅜ nonferrous	Above lath	Expanded metal[c]	¾	none	Average	0.88

[a] Nonferrous tube sizes nominal. Ferrous pipe size I.P.S.
[b] Metal lath was diamond type weighing 3.1 lbs per square yard.
[c] Metal lath was flat rib type weighing 2.5 lbs per square yard.
[d] Nominal thickness of plaster under tubes.
[e] C_e = Downward heat flow divided by tube to surface temperature difference. Values are for 8-in. tube spacings, and a ratio of upward to downward heat flow = 0.2.

The upward heat flow from the panel may be calculated from knowledge of the construction and insulation above the panel and the temperature difference between the panel and some convenient location above the panel. For precise calculations, it would be necessary to use the temperature at the back of the panel. However, this is never known, and for all practical purposes it is sufficient to use the panel surface temperature which is one of the design criteria.

APPLICATION OF RESULTS

The results of these studies will permit the design of a heating panel when the required heat output and the panel surface temperature have been deter-

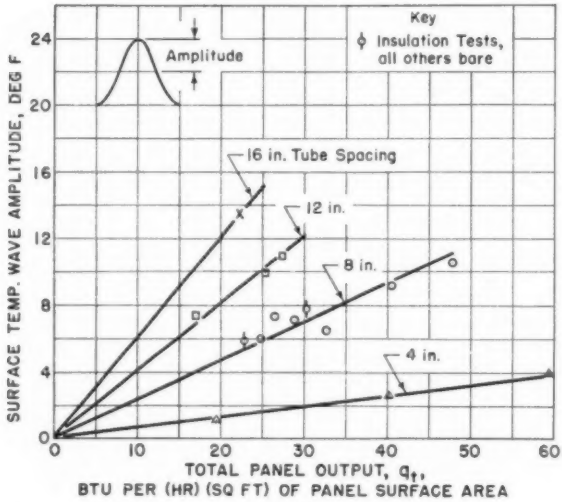


FIG. 7. SURFACE TEMPERATURE WAVE AMPLITUDE VS. PANEL OUTPUT: PANEL NO. 6

mined. The relationship between these two variables must be established for each separate application. Data on the relationship between panel output and panel surface temperature are now being developed at the A.S.H.V.E. Research Laboratory as part of the overall program on panel heating and cooling; one paper, on ceiling panels, has been published.[4]

Examples 1 and 2 illustrate the use of results from the study herein reported.

Example 1: Determine the required tube temperature for a panel which will have a heat output (q_a) of 40 Btu per (hr) (sq ft) with a room temperature of 70 F

and an outside temperature of zero. The overall upward conductance from the panel to the outside air is 0.3 Btu per (hr) (sq ft) (F deg).

For this example it will be assumed that the panel is similar in construction to panel No. 5 of this paper, and that the tubes are on 8-in. centers. It is further assumed that for this particular example, a panel surface temperature of 100 F will be required to give a heat output of 40 Btu per (hr) (sq ft).

The unknowns are the upward heat flow, q_u, and the tube temperature.

Solution: The upward heat flow using the panel surface temperature, 100 F, the upward conductance 0.3 Btu per (hr) (sq ft) (F deg), and the outdoor temperature of 0 F is given by the equation,

$$q_u = C\Delta t = 0.3(100\text{-}0) = 30 \text{ Btu per (hr) (sq ft)}$$

The ratio of upward to downward heat flow is

$$q_u/q_d = 30/40 = 0.75$$

The effective conductance C_e from Fig. 3 = 1.05 Btu per (hr) (sq ft) (F deg). The tube temperature = q_d/C_e + Panel Surface Temperature = 40/1.05 + 100 = 138 F.

Example 2: Determine the required tube temperature if all values remain the same as in *Example 1* except that the overall upward conductance C has been reduced to 0.1 Btu per (hr) (sq ft) (F deg) by adding insulation to the back of the panel.

TABLE 2—ESTIMATED TUBE TEMPERATURES FOR PLASTER PANELS WITHOUT BACK-PLASTERING FOR CONDITIONS GIVEN[a]

PANEL NUMBER	EFFECTIVE PANEL CONDUCTANCE C_e Btu/(hr) (sq ft) (F DEG)		TUBE TEMPERATURE F		
	4 inch Spacing	8 inch Spacing	4 inch Spacing	6 inch Spacing[b]	8 inch Spacing
2	4.13	1.53	109.7	117.9	126.1
5	3.38	1.38	111.8	120.4	129.0
4	3.35	1.29	111.9	121.5	131.0
6	2.66	1.11	115.0	125.5	136.0
3	2.16	0.98	118.5	129.7	140.8
1	2.07	0.86	119.3	132.9	146.5

[a] Panel Surface Temperature 100 F.
Overall Upward Conductance C = 0.1 Btu/(hr) (sq ft) (F deg).
Outside Air Temperature 0 F.
q_u (Upward Heat Flow) = C (Panel Surface Temp. − Outside Air Temp.) = 0.1 (100 − 0) = 10 Btu per (hr) (sq ft).
q_d (Downward Heat Flow) = 40 Btu per (hr) (sq ft).
q_u/q_d (Ratio of Upward to Downward Heat Flow) = 10/40 = 0.25.
None of panels back-plastered.
[b] Linear interpolation of tube temperatures.

The upward heat flow is 0.1 (100−0) = 10 Btu per (hr) (sq ft). From Fig. 3 the effective conductance C_e is 1.38 Btu per (hr) (sq ft) (F deg). The required tube temperature is 40/1.38 + 100 = 129 F.

Table 2 shows the required tube temperatures for each of the six panels as calculated for conditions of *Example 2*. It may be seen that the required tube temperatures are approximately the same for the three types of well constructed panels, Nos. 2, 5, and 4. The increase in tube temperature for panel No. 6 is caused by poor tube embedment. Although panels Nos. 3 and 1 had average tube embedment, the lath and tubes were not tied together, and consequently a considerably higher tube temperature is indicated.

The limitations of maximum panel surface temperatures and water temperatures as given in the technical press emphasize the importance of proper panel design and construction.

CONCLUSIONS

1. The heat transfer within a plaster panel is related to the heat output from the lower and upper surfaces of the panel and can be expressed in terms of an effective conductance. This is defined as the downward heat flow in Btu per (hour) (square foot) divided by the difference between the average tube temperature and the average panel surface temperature in Fahrenheit degrees.

2. For all practical purposes, panel surface temperatures may be used for calculation of the upward heat flow.

3. Insulation on the back of plaster panels serves the following purposes: (*a*) reduces upward heat flow; (*b*) increases the average panel surface temperature for a given tube temperature, or conversely; (*c*) for a given downward heat flow, permits operation with a lower tube temperature than would be required if the panel were not insulated.

4. The addition of back-plastering to panels constructed with tubes above metal lath increases the heat transfer to the panel surface. In general, the heat transfer of a panel with tubes on 6-in. centers without back-plastering is equivalent to that of a panel with tubes on 8-in. centers with back-plastering.

5. It has been demonstrated that for panels having tubes above metal lath, good tube embedment and good contact between tubes and lath are prerequisite to good heat transfer.

REFERENCES

1. A.S.H.V.E. RESEARCH REPORT No. 1426—Laboratory Studies of the Thermal Characteristics of Plaster Panels, by C. M. Humphreys, C. V. Franks, and L. F. Schutrum (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 363).

2. Standard Specifications for Gypsum Plastering A42.1-1946, obtainable from *The American Society for Testing Materials* or *The American Standards Association.*

3. A.S.H.V.E. RESEARCH REPORT No. 1453—A Low-Inertia Low-Resistance Heat Flow Meter, by R. G. Huebscher, L. F. Schutrum, and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952. p. 275).

4. A.S.H.V.E. RESEARCH REPORT No. 1473—Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 197).

DISCUSSION

C. F. KAYAN, New York, N. Y. (WRITTEN): Continued progress in the understanding of panel performance is again evidenced by this contribution from the Research Laboratory. Particularly important are items 4 and 5 of the conclusions.

The thermal behavior of these panels is directly concerned with the communication of the tube temperature to the panel surface, in which metal lath, plus plaster, and their *contact* with the tube are most important. The contact element at the tube seems to have the most outstanding effect, hence here we might expect that the quality of a job would depend on the effectiveness of such tube contact. Back-plastering and good tube embedment, plus metallic contact by means of wiring, would seem to be most essential to the extent that perhaps wiring spacing closer than 8 in. might be desirable.

It would also have been interesting if back-plastering had been applied to the No. 5 panel rather than just to the No. 6 panel, especially since Table 1 indicates that the No. 6 panel tube embedment was considered poor.

The inclusion of the examples of computation in the paper has been most helpful in interpretation and the utilization of the results.

D. L. MILLS, Rome, N. Y. (WRITTEN): This excellent paper may be considered as an evidence of the responsibility that the members of the Technical Advisory Committee on Panel Heating and Cooling feel in the development and publication of useful data. The work outlined was not undertaken until after the original four panels had been tested and the results published. It is our belief that the information published in this paper fully justifies the additional work involved and thanks are due to the laboratory staff for the excellent cooperation they have given the committee on this project.

There are two mechanisms involved in the overall heat transfer from the heating medium to the enclosure. The first involves the heat transfer within the panel itself and the second, the energy release from the panel surface. This paper clearly shows the variations in heat transfer characteristics *within the panel* that may result from different installation conditions. That was the primary objective in the minds of some members of the committee when suggesting these additional tests. This information is so important that the paper should be studied by designers and contractors responsible for panel heating installations. The importance of good contact between the heating coil and the metal lath is emphasized.

Note, for example, from Table 2 that in the case of 6 in. tube spacing, which is the most common for ceiling panels, an increase of 12.5 deg in *mean water temperature* is necessary in order for a *poorly* constructed panel to produce the same rate of heat emission as a panel of *good* construction operating under simliar environmental conditions.

Table 2 also substantiates the results of work done at the University of California in 1941-42 and indicates that an *overall* correlation of panel output in terms of water-to-room air temperature difference is irrational and unlikely to lead to dependable coefficients.

Attention is further directed to the relation between tube spacing in the panel and mean water temperature in the coil, required to produce a given rate of heat emission under given conditions for all six of the panels tested. This relation frequently seems to be lost sight of in panel design for radiant heating systems.

The authors have used the term *tube temperature*. We assume that to be substantially the same as the *mean water temperature* in the coils, which is the term used in

these comments and which we believe to be the more common in design discussions. We would like to know if this assumption is correct?

E. L. SARTAIN, Urbana, Ill. (WRITTEN): In the conclusions it is stated that the effect of insulation on the back of plaster panels serves the following purposes: (1) reduces upward heat flow; (2) increases the average panel surface temperature for a given tube temperature, or conversely; (3) for a given downward heat flow, permits operation with a lower tube temperature than would be required if the panel were not insulated. These conclusions seem logical and would be expected under normal operation, but according to Figs. 5 and 6 insulation has no effect on the surface temperature wave amplitude in the tests when compared on the basis of total panel output.

What could be expected in the variation of surface temperature wave amplitude when plotted against the ratio of upward to downward heat flow, or when plotted against downward heat flow rather than total heat flow for the cases—including insulation, and without insulation? In other words, under normal conditions of operation (in the field) it would seem logical that the effect of the insulation on the reverse side of the panel would tend to increase the mean panel surface temperature for a given tube temperature, thereby causing a larger heat flow downward with insulation than without at the same tube temperature.

From the work of Hulbert, Nottage, and Franks reported in a previous paper*, the theory developed indicates that the heat flow through a panel is made up of two components: source heat flow + linear heat flow. By keeping the source heat flow constant and increasing the linear heat flow, the total output would be increased. It seems that the use of insulation on the reverse side of the panel would cause an increase in temperature on the reverse surface thereby increasing the linear component of heat flow for the same tube temperature. This downward heat flow is also a function of the absorber temperature because of the effect of the absorber temperature on the panel surface temperature. For this reason, I am interested in knowing the range of absorber temperatures during the tests.

These remarks are made in the interest of relating the many variables in the entire panel heating system to the performance of the panel; *i.e.*, environmental conditions, reverse insulation, tube temperature, etc. Also, the relationship between the theory developed in the previously mentioned paper and the test results presented here are not clear in my mind. I wonder whether the theory of heat flow developed for concrete floor panels can be applied to ceiling panels since the embedding of the tubes is an additional variable to be considered.

W. M. WALLACE, II, Durham, N. C.: Table 1 shows ½-in. ferrous tubing and ⅜-in. nonferrous tubing. It was not pointed out that these two different tubes had a different surface area. I wonder if the authors have any information at this time that would indicate whether the actual diameter or surface area has any large effect on the output of the panel?

R. L. MAHER, Waterbury, Conn.: It has taken two years and two papers to arrive at some data wherein we have only changed the variable of the actual mechanics of construction of these panels. When we started with radiant heating we were concerned in the field with too many variables. For that reason panel heating was brought into the laboratory where the variables could be controlled. However, in order to simulate field conditions it was necessary to reintroduce the variables.

---

* A.S.H.V.E. RESEARCH REPORT No. 1388—Heat Flow Analysis in Panel Heating or Cooling Sections: Case I—Uniformly Spaced Pipes Buried Within a Solid Slab, by L. E. Hulbert, H. B. Nottage, and C. V. Franks (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 189).

Some of the considerations in panel construction are types of lath, plaster, embedment and workmanship, all of which affect performance.

We should avoid being too specific and limiting the application of studies. We have in the lath, for example, everything from the standard expanded metal right down to chicken wire, which will give different results. We have to remember that the people who may be installing these panels, doing the plastering, and putting in the lath are not heating engineers and may not do a workmanlike job. Panels must be designed so there will be good contact between the tube and plaster.

I agree with conclusion 3 of the paper in principle, but I question whether insulation by itself will improve the performance of a ceiling panel below a heated space. Introducing insulation above the panel will force only a small portion of the heat down. The heat cannot be driven and that conclusion should not be made from the paper.

P. B. GORDON, New York, N. Y.: The results of the studies described in this paper have certainly justified the research expense. They demonstrate another example of the importance of checking and rechecking where certain data are questioned and further tests are necessary to indicate the differences of various constructions. We can now establish an average value for average construction, and predict the upper and lower limits due to such differences.

I disagree with Mr. Maher about the need for insulation even if the space above is heated. I believe that such insulation forms an important part of the panel heating mechanism of the ceiling panel construction. The data shown in this paper help to substantiate this point.

Further, all of this ties into some of the other comments on the earlier paper that were made on the problem of responsibility of construction. The installation features of a job should be as much the responsibility of the people who design the job as it is of those who install the job. They may not have the true legal responsibility, that is they may not have the responsibility delegated to them, but somewhere along the line that responsibility should be assumed by the designer. A system in itself or the mechanism should not be banned because someone down the line may have installed it improperly. I believe that specifications should be set up and requirements established and then someone be obligated to meet these requirements.

L. A. BURCH, Detroit, Mich.: A few years ago we were told by a manufacturer of gypsum products that temperatures in the neighborhood of 125 to 130 deg were maximum for plaster to avoid difficulty from calcination. This paper described temperatures as high as 146.5 deg. Has the plastering industry raised the maximum temperature limits that were given us a few years ago?

A source of trouble on plaster panel jobs that we have frequently encountered is caused perhaps by a lack of coordination and understanding between the architectural and the mechanical engineers and workmen on a particular job. We have been called in on several well designed plaster panel jobs and have discovered that the architect has specified or allowed the use of vermiculite plaster. As a result the heating system has almost always failed to operate satisfactorily. How much will vermiculite plaster cut down the effectiveness of a ceiling panel? It must be a tremendous amount from the results that I have observed.

E. L. WEBER, Seattle, Wash.: I have designed 180 panel heating jobs in the past 15 years and have made observations from the practical engineering standpoint.

I will not use copper coils because the copper expands about 20 percent more than concrete and plaster, and steel expands 15 percent less. I have seen some copper coil installations where the copper broke away from the plaster. This produces an air

space between the coil and the plaster and greatly reduces the transmission of heat from the coils to the panel surfaces.

In some copper floor coil installations, I have seen numerous cracks, or even chasms in the concrete floor, while we have had some steel floor coils with loops over 30 ft long and no cracks appeared in the concrete floor.

The determination of the safe maximum length of loops, which can be used in plaster or concrete with varying temperatures of water for both copper and steel pipe, before cracks appear in the panel surface, offers a valuable field of experimentation. The small panels used in the laboratory will not provide any such valuable data.

In the illustration of the test panels the metal coils apparently were fastened directly to the joists which is poor field practice, because some joists will warp downward and some upward. In one job a contractor fastened the coils to the joists in place of using the supporting pipe battens shown on the plans. The ceiling broke up and came down shortly after being put in operation.

In practice it is difficult to back plaster in the usually limited space available. Plaster must invariably be applied from the finished side of the plaster panel. By allowing the steel pipe coils to lie in the open and permitting a slight coating of rust to replace the usually present external oil film, good results have been obtained. The coils and metal lath are then wetted before applying the first coat of plaster. As the plaster is pushed through the lath, capillary action will draw the plaster around the coils. The wetting will also avoid withdrawing the moisture from the plaster, thus preventing many fine hair cracks from forming.

In practice batts are almost impossible to install, due to the presence of hangers, etc. They also produce void spaces between the batts and the undulating plaster key surfaces over the coils. Air circulation can occur in these voids and greatly affect heat output of panels, especially in coved and slanting ceilings. Tests should be conducted using some of the more practical materials which can be *blown in* behind the panel surfaces and which also eliminate voids; such as loose mineral wool, vermiculite, etc.

In ceiling panels in concrete slabs, we formerly placed the coils on the forms and applied three-coat plaster ceilings, but this was often too expensive. In a number of buildings we have recently used plywood forms and set the coils on transite chips—$2 \times 1 \times \frac{1}{8}$ in. at 36 in. centers. This was found to completely bury the coils and the ceilings were finished with a skim coat of plaster, or even paint or enamel. If this method is used, the transite slabs should not be cut with a saw but broken in a sheet metal break.

In a number of illustrations the design consists of a concrete sub-slab upon which the coils are laid, and over this is poured a rather thin concrete finish slab. This makes a very risky installation, as there is produced a definite cleavage plane, as in slate or shale. When the coils are quickly heated, their expansion may cause the finish slab to be buckled upward. It is preferable to pour the slab monolithic, above and below the coils at the same time. For this purpose the coils can be set on pipe stands, or on concrete sleepers about 3 ft on centers, and the entire slab can be poured at one time.

Due to the great difference in the thermal coefficients of expansion of steel, copper and plaster, the laboratory setup is risky for field installations. Possibly in the small 6 ft square panel employed they will hang together, but in the much larger panels used in practice, displacement and breaking of contact may result. Steel metal lath should be used with steel pipe and copper lath with copper pipe.

It is very important to avoid the breaking of contact between the pipe, metal lath and plaster, or concrete. Such breaks produce a film of insulating air between the coil and the panel surface, which greatly reduces the conduction of heat.

In one installation with which I am acquainted the power was off during the night for several hours. In the morning the building manager set the temperature limiting thermostat from 135 to 180 F. A number of the steel pipe coils *came loose* in their contact with the panel surfaces. In order to properly heat this building now it is necessary to use water at a temperature 15 to 20 deg higher than for which originally designed. When heat is turned on the movement of the coils in the plaster ceiling produces a noise similar to the chirp of an army of crickets.

I bring up these items to stress the fact that experiments carried out with a 6 ft square panel may give results which would not be applicable in larger panels and would fail to show critical conditions which would produce cracks in the larger panels.

AUTHORS' CLOSURE (C. M. Humphreys): In reply to Mr. Burch's question concerning the temperature limitations for plaster panels recommended by the Gypsum Association, it should be noted that the limits have not been raised; but instead have been lowered to a maximum water temperature of 125 F. The 146.5 F temperature given in Table 2 was for a panel of poor construction and therefore illustrates the importance of good panel construction in limiting the tube temperature to safe values.

In research it is frequently desirable to make tests at conditions beyond those usually used in practice. The fact that tests were made with water at 146.5 F does not indicate that such a temperature is recommended for actual use.

The effect of insulating plasters on panel performance has not been determined in the laboratory; however, a reasonable estimate can be made, neglecting the effect of the lath, on the basis of plaster conductivity alone. It appears that the resistance to heat flow in a panel using insulating plaster mixes may be 2 to 4 times as high as one using the standard gypsum and sand mix. The ensuing temperature difference between the water and the panel surface undoubtedly would be excessive for all but relatively small panel outputs.

Mr. Maher and Mr. Sartain have discussed the ramifications of conclusion 3 in the paper concerning the effect of insulation on the panel output. Insulation on the reverse side of the panel does increase the mean panel surface temperature for a given tube temperature, thereby causing a larger flow downward with insulation than without. This effect is shown for panel No. 1 in Fig. 8 of Reference 1. It is also indicated by the fact that the *effective conductance* of a panel is greater at the lower ratios of upward to downward heat flows than at the higher values.

Mr. Sartain has considered the theory of a source heat flow plus linear heat flow as developed by Hulbert, Nottage and Franks in their paper*. The present form of solution to this theory involves certain limitations, such as a homogeneous media and isothermal bounding surfaces to which the plaster panels do not conform. Thus, a prediction of plaster panel performance could not be made from this theory without extending it to cover other conditions. However, the performance of a plaster panel, as described in the preceding paragraph and as discussed by Mr. Sartain, conforms to the general trends indicated by the theory.

In the third paragraph of Mr. Sartain's discussion, we have interpreted the term *total output* to mean the sum of the source and linear heat flows to the under side of the ceiling panel. In the paper by Hulbert, Nottage, and Franks, the term *total output* was used to describe the sum of the heat flows to the two faces of the panel. This is mentioned only to avoid any possible confusion.

In answer to Mr. Wallace's question, no attempt was made to determine the effect of tube diameter or tube surface area on the panel performance. However, the first

* A.S.H.V.E. RESEARCH REPORT No. 1388—Heat Flow Analysis in Panel Heating or Cooling Sections; Case I—Uniformly Spaced Pipes Buried Within a Solid Slab, by L. E. Hulbert, H. B. Nottage, and C. V. Franks (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 189).

paper on this subject pointed out that the tube surface area in contact with the plaster was about the same for panel No. 3, using ½-in. ferrous pipe, as for panel No. 1, using ⅜-in. nominal nonferrous tubing, and that the performance of these two panels was essentially the same.

In answer to Mr. Mills, the tube temperatures referred to in the text are substantially the same as the mean water temperatures. Actually, the temperature measured was that of the tubes; however, the water velocities were sufficiently high to make the difference insignificant.

1492

No. 1492

HEAT FLOW ANALYSIS IN PANEL HEATING OR COOLING SECTIONS

Case II—Floor Slab on Earth with Uniformly Spaced Pipes or Tubes at the Slab-Earth Interface

By H. B. NOTTAGE†, C. V. FRANKS*, L. E. HULBERT†† AND L. F. SCHUTRUM**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

A PREVIOUS paper[1] dealt with the case of a uniformly-spaced row of pipes contained in a slab having both surfaces isothermal. This paper presents a second case which is typified by a floor slab on the earth, considered to have an isothermal upper surface and formed by concrete poured over pipes laid on the ground.

These investigations are part of the long-range program in panel heating and cooling at the A.S.H.V.E. Research Laboratory. The studies have been conducted under advisory guidance of Group A$^{\Delta}$ of the 1952 Technical Advisory Committee on Panel Heating and Cooling. This group is concerned with problems of heat flow within and behind the heating or cooling section.

The results presented herein have been attained by a combination of mathematical analysis and electrolytic-analogue studies.

SCOPE

Thermal resistances for heat transfer between the pipe surface and the upper (isothermal) slab surface have been established, for the conditions summarized in Fig. 1, for pipes within the slab and tangent to the slab-earth

† Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

* Research Fellow, A.S.H.V.E. Research Laboratory.

†† Graduate Assistant in Mathematics, Case Institute of Technology.

** Research Fellow, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.

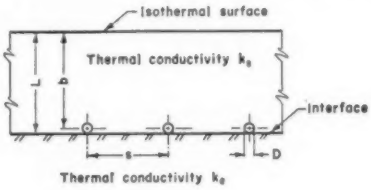
[1] Exponent numerals refer to References.

$^{\Delta}$ *Personnel—Group A:* R. S. Dill, *Chairman;* H. L. Flodin, F. E. Giesecke, G. D. Lain, R. L. Maher, V. L. Miller, D. L. Mills, C. W. Nessell, F. E. Parsons, S. I. Rottmayer, R. T. Schoerner, E. E. Scott, S. K. Smith, R. K. Thulman, S. M. Van Kirk, L. H. Yeager.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Denver, Colo., June 29-30, July 1, 1953.

interface. It must be emphasized that in this analysis *all* heat from the pipe surfaces flows through the upper (isothermal) surface, either directly through the slab from pipe to surface, or into the ground and then up between the pipes to the slab surface. This is a consequence of the assumption made in the analysis that the slab is infinite in extent and therefore there can be no *net* heat flow into the earth. This combination of isothermal upper slab surface and *zero heat loss to the earth* is *never* fully realized in practice but is a practical limiting condition in that it represents the best panel performance theoretically possible.

Fig. 2 repeats the range of geometrical proportions covered in the previous reports[1,2] on slabs having both surfaces isothermal. The range of conditions



Range of conductivity ratios

$\frac{k_s}{k_e}$ = 0.60 and 1.16, all cases

Range of geometrical proportions

$\frac{s}{L}$ = 0.8 to 2.7 at $\frac{D}{L}$ = 0.0625

$\frac{s}{L}$ = 0.8 to 2.7 at $\frac{D}{L}$ = 0.125

$\frac{s}{L}$ = 1.6 to 4.0 at $\frac{D}{L}$ = 0.250

$\frac{s}{L}$ = 2.7 to 5.0 at $\frac{D}{L}$ = 0.313

FIG. 1. RANGE OF CONDITIONS FOR THE THERMAL-RESISTANCE DATA OF THIS PAPER FOR PIPES WITHIN THE SLAB AND TANGENT TO THE SLAB-EARTH SURFACE

between those described in Figs. 1 and 2 is covered by the general mathematical analysis of the present paper.

For the purpose of checking the electrolytic analogue, limited calculations were made for the pipes located with their centers in the interface as noted in Fig. 3. This condition was selected for calculation because the equation for this case is less involved.

The influence of varying the ratio of the slab thermal conductivity to the earth thermal conductivity, beyond the conditions noted in Fig. 1, was investigated for a single geometrical condition as indicated in Fig. 4.

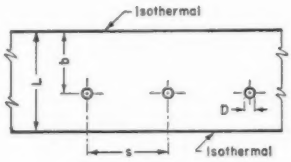
The mathematical analysis is given in detail in the Appendix. This analysis is fundamentally complete, and includes the following conditions beyond the present case: (1) pipe locations intermediate between those shown in Figs. 1 and 2, and (2) a single source in a slab having surfaces non-isothermal. The case of a row of pipes in a slab with both surfaces non-isothermal may be

derived from the latter conditions. Much work is required to express these further general mathematical solutions in simple terms for practical application calculation but this important aspect is beyond the present intent.

All results and considerations are restricted to steady-state conditions. Viewing practical applications of the data, the earth and the slab must have uniform thermal conductivities and the regions of *edge effects* are to be excluded from the slab.

MATHEMATICAL ANALYSIS

The mathematical derivation is an extension of the previously presented case[1], employing the method of a source and image system. Treatment of a non-isothermal boundary interface involves two considerations: (1) that the temperature of the interface at any point must be the same when approached from either side, and (2) that the rate of heat flow at the interface must be the same when approached from either side. These lead to establishing the ratio of the thermal conductivities of the two adjacent media as an important variable.



$\frac{b}{L}$	$\frac{D}{L}$	$\frac{s}{L}$
0.15	0.10	2.0
0.30	0.30	--
0.50	0.80	6.0

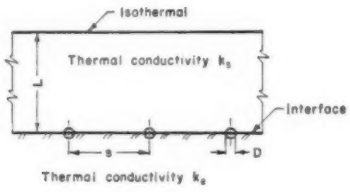
FIG. 2. RANGE OF CONDITIONS FOR THE PREVIOUS CASE (SEE REF. 1) OF PIPES WITHIN A SLAB HAVING BOTH SURFACES ISOTHERMAL

The isothermal boundary solution is obtained from this general case by letting the thermal conductivity of the medium adjacent to the slab (in this case, the air) approach an infinite magnitude. The situation of a non-isothermal slab boundary with an adjacent fluid medium (air) can also be handled through the introduction of the thermal conductance between the surface and the fluid body.

The thermal resistance, R, of interest is that between the pipe surface and the isothermal slab surface; it is defined as

$$R = \frac{\text{(Temperature difference between pipe surface and isothermal slab surface)}}{\text{(Mean rate of heat flow per unit area of the isothermal slab surface)}}$$

The system envisioned is diagrammed in Fig. 1.



For $\frac{k_s}{k_e} = \infty$

$\frac{s}{L} = 0.5$ to 4.0 at $\frac{D}{L} = 0.0625$

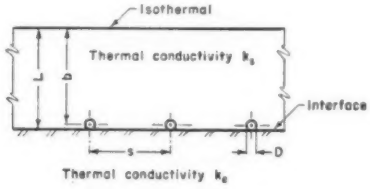
For $\frac{k_s}{k_e} = 1$

$\frac{s}{L} = 0.5$ to 4.0 at $\frac{D}{L} = 0.0625$

FIG. 3. CONDITIONS EVALUATED FOR THE PIPES BEING LOCATED WITH THEIR CENTERS IN THE SLAB-EARTH INTERFACE

ANALOGUE INVESTIGATIONS

The final mathematical equation‡ for the thermal resistance, R, is very difficult to evaluate without the aid of special computing devices. In order to focus upon practical application results, two compromises were made: (1) that only a limited range of geometrical proportions and thermal conductivities would be considered, these being representative of panel heating floor slabs on the earth, and (2) that a certain inaccuracy would be tolerated in the results, since the practical application did not warrant high precision.



For $\frac{s}{L} = 5.0$ and $\frac{D}{L} = 0.313$

$\frac{k_s}{k_e} = 0$ to 5.0

FIG. 4. CONDITIONS COVERED IN EXPLORING THE INFLUENCE OF EARTH AND SLAB THERMAL CONDUCTIVITIES

‡ This equation, because of its complexity, is not included here but is given in full in the Appendix.

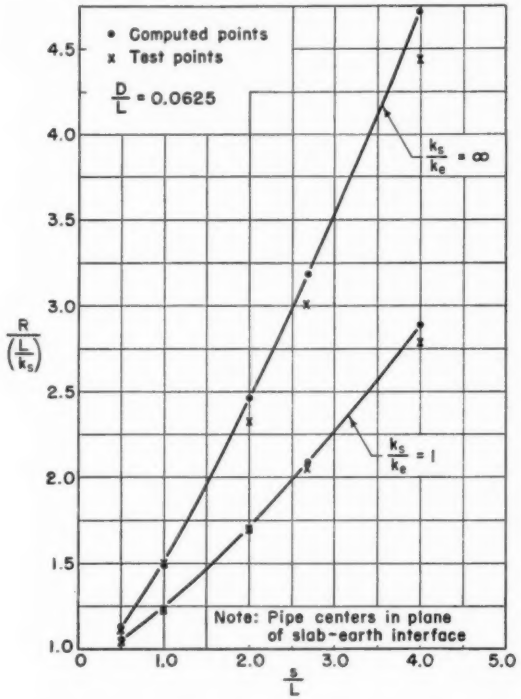


FIG. 5. COMPARISON OF COMPUTED RESULTS AND ANALOGUE DATA FOR THE SYSTEM OF FIG. 3

The following steps were then taken:

1. The equation for the thermal resistance was evaluated for the range noted in Fig. 3, with the result shown in Fig. 5.

2. The electrolytic analogue, as employed previously[2], was set up to solve the same problem. The comparison between the mathematical results and the analogue data is given in Fig. 5. This serves as an indication of the analogue accuracy to be expected. In using the analogue, different thermal conductivities were simulated through different depths of electrolyte on the opposite sides of the interface.

3. Judging the analogue to be acceptable, measurements of thermal resistance were made over the range in Fig. 1. The results are shown in Fig. 6, wherein the $R/(L/k_s)$ ordinate is the ratio of the measured resistance to the resistance of a simple slab having isothermal surfaces and with the same thickness and thermal conductivity as the slab concerned.

4. A supplementary test was made on the effect of variations of thermal conductivity, as noted in Fig. 4 and shown in Fig. 7.

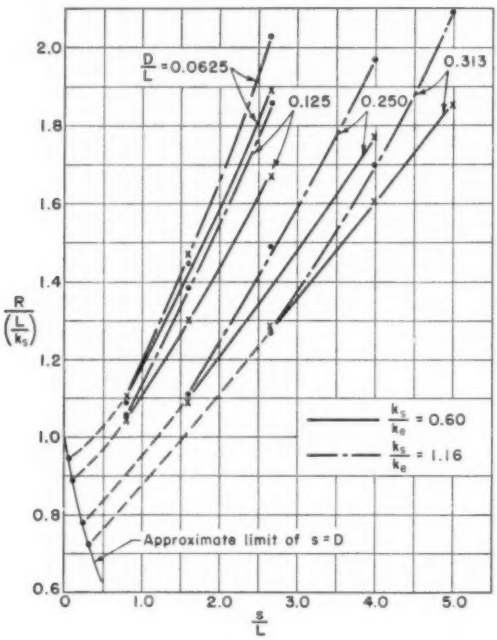


FIG. 6. THERMAL RESISTANCE MAGNITUDES FOR THE PRACTICAL RANGE OF SLAB GEOMETRIES GIVEN IN FIG. 1

DISCUSSION

Table 1, column 4, gives the slab resistance for some practical values of tube size, tube spacing, and slab thickness which are covered by the general dimensionless ratios of Fig. 6. The effect of varying the slab thickness, tube size, and tube spacing upon the slab resistance is shown for any heat flow rate or slab surface temperature. The tube temperatures shown in Table 1 are the minimum temperatures which would produce an 85 F isothermal upper surface temperature and a surface heat flow rate of 40 Btu per (hr) (sq ft) with the specified slab and earth conductivities. *In practice, tube temperatures will always be higher than those illustrated* and will be highest for wider tube spacings which produce the greatest departure from isothermal surface conditions.

The effect of earth conductivity upon the values given in Table 1 can best be shown from Fig. 7. For a decrease in earth conductivity, k_e, the ratio k_s/k_e becomes larger, resulting in a higher slab resistance. In practice, the higher slab resistance resulting from the lower earth conductivity would be somewhat compensated by a reduced heat loss into the earth.

The resistance magnitudes presented cover only a limited range pertinent to most common usage in panel heating. The mathematical analysis, however, is

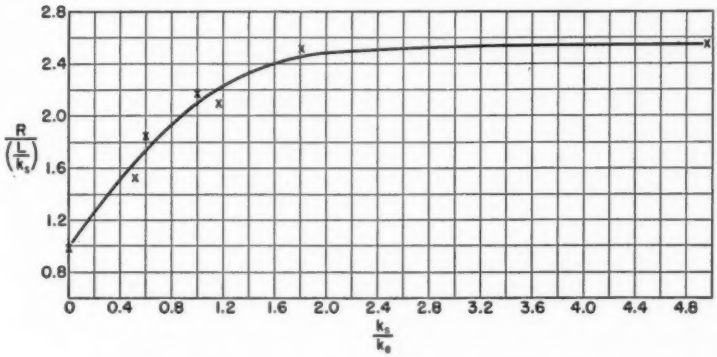


FIG. 7. ILLUSTRATIVE EFFECT OF SLAB-EARTH CONDUCTIVITY RATIO k_s/k_e UPON THE THERMAL RESISTANCE FOR THE SYSTEM OF FIG. 4

perfectly general and may be applied to other than panel heating problems. A heat pump source located at a distinct boundary between two media is one example of a related system.

The steady-state limitation must be kept in mind. While it may be common practice to base design estimates on steady-state conditions, it is very doubtful if such ever prevail, strictly speaking, in actual panel-heating operation.

The thermal resistance reported in this paper is but one element of a complete thermal circuit. For a floor slab, one branch of this circuit would go into the

TABLE 1—TUBE TEMPERATURES FOR 85 F SLAB SURFACE TEMPERATURE AND 40 BTU PER (HR) (SQ FT) SLAB HEAT OUTPUT

SLAB THICKNESS L IN.	TUBE SIZE D IN. O.D.	TUBE SPACING S IN.	SLAB RESISTANCE[a] (HR) (SQ FT) F DEG/BTU	TUBE TO SURFACE TEMP DIFF F DEG	TUBE TEMP F
4	5/8	4	0.363	14.5	99.5
4	5/8	8	0.492	19.7	104.7
4	5/8	12	0.630[b]	25.2	110.2
6	5/8	4	0.515	20.6	105.6
6	5/8	8	0.645	25.8	110.8
6[c]	5/8	12	0.790	31.6	116.6
6	5/8	16	0.960[b]	38.4	123.4
6	1 3/8	4	0.452	18.1	103.1
6	1 3/8	8	0.532	21.3	106.3
6	1 3/8	12	0.650	26.0	111.0
6	1 3/8	16	0.772	30.9	115.9

[a] Slab resistances (tube to isothermal surface) for $k_s = 12.0$ and $k_e = 10.35$ Btu/(hr) (sq ft) (F deg per in.)
[b] Estimated by extrapolation of curves in Fig. 6.
[c] Conditions for illustrative problem.

earth, and the other branch would run out through the heated structure. Studies of this circuit are an important problem deserving attention.

Cases of non-uniform pipe spacing may be dealt with by an extension of the theory or by using the analogue. Consideration of the thermal conditions near the edge of a finite slab also is possible by the theory but would involve very complicated detail; the analogue is available as a more simple approach.

The results in Fig. 6 have not been checked by mathematical computations. Rough judgment of the accuracy is available from Fig. 5.

CONCLUSIONS

1. The practical case of a floor slab on the earth, for the limiting conditions of zero heat loss to the earth and isothermal upper slab surface, has been solved in complete mathematical form, demonstrating the utility of the method of mathematical analysis invoked.

2. The final equation from the theory was found to be too complicated for general numerical work; analogue data were therefore obtained over the limited range of panel heating interest.

ILLUSTRATIVE PROBLEM

A. Determine the thermal resistance for the following concrete slab:

Slab
Thickness $L = 6$ in.
Tube Spacing $s = 12$ in.
Tube Size $D = \frac{5}{8}$ in. ($\frac{1}{2}$ in. nominal)
Slab
Conductivity $k_s = 12$ Btu per (hr) (sq ft) (F deg per in.).
Earth
Conductivity $k_e = 10.35$ Btu per (hr) (sq ft) (F deg per in.).
Using the given values,

$$s/L = 2, \quad D/L = 0.104, \quad k_s/k_e = 1.16.$$

From Fig. 6 $R/L/k_s = 1.58$ by interpolation between $D/L = 0.0625$ and $D/L = 0.125$. Substituting for L and k_s, $R = 0.79$ (hr) (sq ft) (F deg) per Btu.

B. Determine the tube temperature in the foregoing problem for a surface heat flow rate of 40 Btu per (hr) (sq ft) and a surface temperature of 85 F.

Tube to surface temperature difference = 40 × *resistance* from tube to surface = 40 × 0.79 = 31.6 F deg.

The tube temperature, therefore, is 116.6 F which is the sum of the surface temperature and the tube to surface temperature differential.

NOMENCLATURE

b = distance between pipe axis and isothermal slab surface, inches or feet.
D = pipe outer diameter, inches or feet.
k_s = thermal conductivity of slab material, Btu per (hour) (square foot) (Fahrenheit degree per inch).
k_e = thermal conductivity of the adjacent medium (*e.g.*, the earth) Btu per (hour) (square foot) (Fahrenheit degree per inch).
L = slab thickness, inches or feet.
R = thermal resistance between pipe surface and opposite isothermal slab surface, defined in the text, (hour) (square foot) (Fahrenheit degree) per Btu.
s = pipe center-to-center spacing, inches or feet.

REFERENCES

1. A.S.H.V.E. RESEARCH REPORT No. 1388—Heat Flow Analysis in Panel Heating or Cooling Sections, Case I—Uniformly Spaced Pipes Buried Within a Solid Slab, by L. E. Hulbert, H. B. Nottage, and C. V. Franks (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 189).

2. A.S.H.V.E. RESEARCH REPORT No. 1387—Laboratory Studies on Heat Flow Within a Concrete Panel, by C. M. Humphreys, H. B. Nottage, C. V. Franks, R. G. Huebscher, L. F. Schutrum and D. W. Locklin (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 175).

APPENDIX

INTRODUCTION

A previous paper[1] presented the analytical solution for Case I in this series, which was a slab containing uniformly-spaced line sources under conditions such that both slab surfaces could be treated as isothermal planes. Case II deals with the steady-state heat flow solution for an infinite floor slab where only the upper surface can be treated as an isothermal plane. Additional cases may be developed from the mathematical theory presented, but these remain for future studies to evaluate them in terms suitable for engineering calculations.

The following mathematical developments are included in the present analysis:

1. The temperature field for a single line source within a semi-infinite medium having a common boundary with a second semi-infinite medium.
2. The temperature field for a single line source in a medium bounded by two parallel plane-surface interfaces with semi-infinite adjacent media.
3. The temperature field for a row of uniformly spaced line sources in a medium bounded by two parallel planes, one of which is isothermal with the other being a non-isothermal interface with a semi-infinite adjacent medium.
4. The heat flow rates corresponding to system 3.
5. The thermal resistance between a pipe surface and the opposite isothermal slab surface, for a uniformly-spaced row of pipes located with their centers in a slab surface which is common with an adjacent semi-infinite medium.

Problem 1. The Temperature Field for a Single Line Source within a Semi-Infinite Medium Having a Common Boundary with a Second Semi-Infinite Medium

Consider first a single line source of heat in a semi-infinite medium of thermal conductivity k_1 with the source axis parallel to the plane surface boundary between this first medium and a second semi-infinite medium of thermal conductivity k_2. Let the source be located at the coordinate position (o,y_o), as shown in Fig. A-1, and have a strength m Btu per (linear foot) (hour). Steady-state conditions are postulated throughout the analysis.

The development follows the approach of the previous paper[1] in that the thermal field is represented as the resultant effect of a source-image system. This resultant is obtained by superposing the contribution of the source and each image but in this case the boundary conditions at the junction of the two media are not those of an isothermal plane.

Referring to Fig. A-2, assume a general condition whereby the temperature field in medium 1 is the same as if medium 1 alone were present and there was a source of strength m at the point $S'(o,y_o)$ and a source of strength m' at the point $S''(o,y_o)$. For medium 2 assume that the temperature field is the same as if this medium alone were present and there was a source of strength n at the point S.

[1] See Reference 1 of the paper.

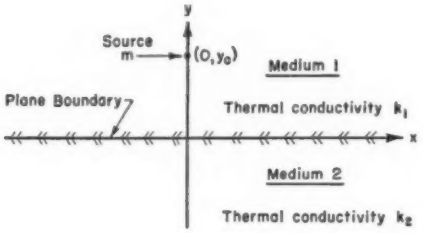


FIG. A-1. COORDINATE SYSTEM AND POSITION OF LINE SOURCE

Take some point P on the boundary surface, $y = 0$. Continuity requirements are:

1. Continuity of heat flow across the boundary.

$$k_1 \left(\frac{\partial \theta_1}{\partial y}\right)_P = k_2 \left(\frac{\partial \theta_2}{\partial y}\right)_P \qquad (A\text{-}1)$$

2. Temperature equality on the boundary.

$$(\theta_1)_P = (\theta_2)_P \qquad (A\text{-}2)$$

The symbol Θ denotes temperature with respect to a datum to be established on the basis of rational convenience. With an isothermal surface, this surface temperature is a useful datum.

For a single source in an infinite medium of thermal conductivity k_1 the temperature at any point is given by the expression derived in Reference 1,

$$\theta_1 = -\frac{m}{4\pi k_1} \log_e r^2 \qquad (A\text{-}3)$$

where r is the distance from the source to the point; see Fig. A-2.

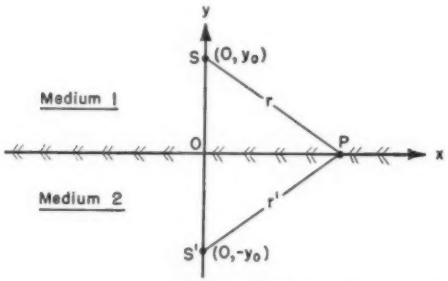


FIG. A-2. INVESTIGATION OF NON-ISOTHERMAL BOUNDARY IN A SEMI-INFINITE MEDIUM

For two sources of strength m and m' respectively in a single infinite medium, the temperature at any point is given by

$$\theta_1 = -\frac{m}{4\pi k_1}\log_e r^2 - \frac{m'}{4\pi k_1}\log_e r'^2 \quad \ldots\ldots\ldots \text{(A-4)}$$

when r' is the distance from the source m' to the point in question.

Following the assumptions noted in the foregoing for the temperature field with two media present, the temperatures $(\theta_1)_P$ and $(\theta_2)_P$ are

$$(\theta_1)_P = -\frac{1}{4\pi k_1}\left[m \log_e r^2 + m' \log_e r'^2\right] \qquad \text{(A-5)}$$

$$(\theta_2)_P = -\frac{1}{4\pi k_2}\left[n \log_e r^2\right] \qquad \text{(A-6)}$$

Further

$$\left(\frac{\partial \theta_1}{\partial y}\right)_P = -\frac{1}{2\pi k_1}\left[\frac{m}{r}\frac{\partial r}{\partial y} + \frac{m'}{r'}\frac{\partial r'}{\partial y}\right] \qquad \text{(A-7)}$$

$$\left(\frac{\partial \theta_2}{\partial y}\right)_P = -\frac{1}{2\pi k_2}\left[\frac{n}{r}\frac{\partial r}{\partial y}\right] \qquad \text{(A-8)}$$

From Fig. A-2, $r = r'$ in magnitude but the sense of the corresponding derivatives with respect to y is opposite; hence,

$$\frac{\partial r}{\partial y} = -\frac{\partial r'}{\partial y}$$

A necessary relationship between the three source strengths follows from substituting the respective quantities above into Equations A-1 and A-2. From Equation A-1,

$$\frac{k_1}{2\pi k_1}\left[\frac{m}{r}\frac{\partial r}{\partial y} + \frac{m'}{r}\left(-\frac{\partial r}{\partial y}\right)\right] = \frac{k_2}{2\pi k_2}\left[\frac{n}{r}\frac{\partial r}{\partial y}\right]$$

$$\therefore\ m - m' = n \qquad \text{(A-9)}$$

From Equation A-2,

$$\frac{1}{4\pi k_1}\left[m \log_e r^2 + m' \log_e r^2\right] = \frac{1}{4\pi k_2}\left[n \log_e r^2\right]$$

$$\therefore\ m + m' = \frac{k_1}{k_2}\, n \qquad \text{(A-10)}$$

Using Equations A-9 and A-10, n may be eliminated and m related to m',

$$m' = -\frac{\left(\frac{k_2}{k_1} - 1\right)}{\left(\frac{k_2}{k_1} + 1\right)}\, m \qquad \text{(A-11)}$$

For simplicity let

$$\mu = \frac{\left(\frac{k_2}{k_1} - 1\right)}{\left(\frac{k_2}{k_1} + 1\right)} \qquad (A\text{-}12)$$

Then,

$$\left.\begin{aligned} m' &= -\mu m \\ n &= (\mu + 1)\, m \\ &= \frac{k_2}{k_1}\, m\,(1 - \mu) \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots \quad (A\text{-}13)$$

This shows that the thermal field about the single source in Fig. A-1 may be represented by the superposed effect of combined sources according to the above conditions.

General expressions for the temperature at any point (x,y) are as follows:

In medium 1,

$$\theta_1 = -\frac{m}{4\pi k_1}\left[\log_e\left[x^2 + (y - y_o)^2\right] - \mu \log_e\left[x^2 + (y + y_o)^2\right]\right] \qquad (A\text{-}14)$$

In medium 2,

$$\theta_2 = -\frac{m(1-\mu)}{4\pi k_1} \log_e\left[x^2 + (y - y_o)^2\right] \qquad (A\text{-}15)$$

The previous solution[1] for an isothermal boundary surface at $y = 0$ may be compared to the present case by placing $\Theta_1 = 0$ and $\Theta_2 = 0$ when $y = 0$ in Equations A-14 and A-15. This establishes the isothermal boundary conditions,

$$\left.\begin{aligned} \mu &= 1 \\ m' &= -m \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots \quad (A\text{-}16)$$

To obtain $\mu = 1$, Equation A-12 requires $k_2/k_1 = \infty$. This could be approached to a good physical approximation by having k_2 very much greater than k_1.

Problem 2. The Temperature Field for a Single Line Source in a Medium Bounded by two Parallel Plane-Surface Interfaces with Semi-Infinite Adjacent Media

Fig. A-3 shows the system for analysis. For the initial part of the derivation, postulate $k_3 = \infty$ so that the upper boundary will be isothermal with the lower boundary non-isothermal. Let all temperatures be referred to the isothermal upper boundary as the datum. The source bury is b and the plane to plane (surface to surface) distance is L. The thermal field will be obtained from an infinite source-image sequence along the y axis, similarly to the case previously analyzed.[1] By the results obtained in the preceding section, the spacing of the images is established as in the previous paper, but with each reflection at a boundary the strength of the image concerned is $(-\mu)$ times the strength of the source or previous image from which it originated. For the upper boundary $y = b$, $\mu = 1$ because of the isothermal condition imposed.

Table A-1 gives the positions and strengths of the first few images. The original source is at the point $x = 0$, $y = 0$. The quantity μ is defined by Equation A-12.

The temperature at any point (x,y) within medium 1 is obtained by superposing the contributions of the original source and all of its images. In such a series each

term will be of the form of Equation 3 with m multiplied by μ raised to some power and with the distance r referred to the locations of each image concerned. Thus,

$$-\frac{4\pi k_1 \theta}{m} = \log_e (x^2+y^2) - \log_e \left[x^2+(y-2b)^2\right] + \mu \log_e \left[x^2+(y+2L)^2\right]$$
$$-\mu \log_e \left[x^2+(y-2b+2L)^2\right] + \cdots\cdots \qquad (A\text{-}17)$$

Combining terms

$$\frac{4\pi k_1 \theta}{m} = \log_e \left[\frac{x^2+(y-2b)^2}{x^2+y^2}\right] + \mu \log_e \left[\frac{\left[x^2+(y-2b+2L)^2\right]\left[x^2+(y-2b-2L)^2\right]}{\left[x^2+(y+2L)^2\right]\left[x^2+(y-2L)^2\right]}\right]$$
$$+\mu^2 \log_e \left[\frac{\left[x^2+(y-2b+4L)^2\right]\left[x^2+(y-2b-4L)^2\right]}{\left[x^2+(y+4L)^2\right]\left[x^2+(y-4L)^2\right]}\right] + \cdots\cdots \qquad (A\text{-}18)$$

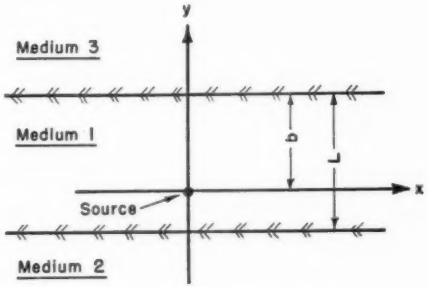


FIG. A-3. A SINGLE SOURCE IN A MEDIUM BOUNDED BY TWO PARALLEL PLANES AND SEMI-INFINITE ADJACENT MEDIA

The series of Equation A-18 is the general solution to the temperature field in the slab represented as medium 1 for the conditions of an isothermal upper boundary.

The condition of both boundaries being non-isothermal is treated in a similar manner, employing superposition. Let

$$\mu_1 = \frac{\frac{k_2}{k_1} - 1}{\frac{k_2}{k_1} + 1} \cdots\cdots\cdots\cdots \qquad (A\text{-}19)$$

$$\mu_2 = \frac{\frac{k_3}{k_1} - 1}{\frac{k_3}{k_1} + 1} \cdots\cdots\cdots\cdots \qquad (A\text{-}20)$$

In this case the source image sequence is established by multiplying the strength of each image's preceding source by $-\mu_1$ for a reflection at the upper boundary where

$y = b$, and by multiplying by $-\mu_2$ for a reflection at the lower boundary, $y = b - L$. Table A-2 gives the positions and strengths of the first few images.

The temperature at any point (x,y) is obtained as before by superposing all image contributions in the form of a series:

$$-\frac{4\pi k_1 \theta}{m} = \log_e (x^2+y^2) - \mu_2 \log_e \left[x^2+(y-2b)^2\right] - \mu_1 \log_e \left[x^2+(y-2b+2L)^2\right] + \mu_1\mu_2 \log_e \left[x^2+(y-2L)^2\right]\left[x^2+(y+2L)^2\right] + \ldots\ldots \qquad (A-21)$$

Equation A-21 is the general solution in mathematical terms. Further manipulation will not be presented at this point because this solution is not the one employed in the case under study. The previous relation, Equation A-18, is obtainable from Equation A-21 by placing $k_3 = \infty$, $\mu_2 = 1$ at $y = b$, which are the requirements for the upper surface being isothermal.

Problem 3. The Temperature Field for a Row of Uniformly-Spaced Line Sources in a Medium Bounded by Two Parallel Planes, One of Which is Isothermal with the Other Being a Non-Isothermal Interface with a Semi-Infinite Adjacent Medium

This problem covers the practical application of a slab on the earth as presented within limits in the body of the paper. The system is sketched in Fig. A-4. The x coordinate of the n^{th} tube or pipe is $x_n = ns$ where n may be positive, negative or zero.

The contribution of each source to the thermal field in medium 1 (the slab) is given by the net effect of its source-image series. Following Equation A-18 the temperature at any point (x,y) in medium 1 and referred to the isothermal surface as a datum, is given by

$$\frac{4\pi k_1 \theta}{m} = \sum_{n=-\infty}^{n=+\infty} \left\{ \log_e \left[\frac{(x-ns)^2+(y-2b)^2}{(x-ns)^2+y^2}\right] + \sum_{K=1}^{K=+\infty} \mu^K \log_e \left[\frac{\left[(x-ns)^2+(y-2b+2KL)^2\right]\left[(x-ns)^2+(y-2b-2KL)^2\right]}{\left[(x-ns)^2+(y+2KL)^2\right]\left[(x-ns)^2+(y-2KL)^2\right]}\right] \right\} \qquad (A-22)$$

TABLE A-1—POSITIONS AND STRENGTHS OF IMAGES FOR THE SYSTEM OF FIG. A-3
With the Plane $y = b$ Being Isothermal

Y	STRENGTH
0 (Source)	$+ m$
$2b$	$- m$
$2L$	$+ \mu m$
$2L + 2b$	$- \mu m$
$4L$	$+\mu^2 m$
$4L + 2b$	$-\mu^2 m$
$6L$	$+\mu^3 m$
•	•
•	•
$2b - 2L$	$- \mu m$
$-2L$	$+ \mu m$
$2b - 4L$	$-\mu^2 m$
$-4L$	$+\mu^3 m$
$2b - 6L$	$-\mu^3 m$
$-6L$	$+\mu^3 m$
•	•
•	•

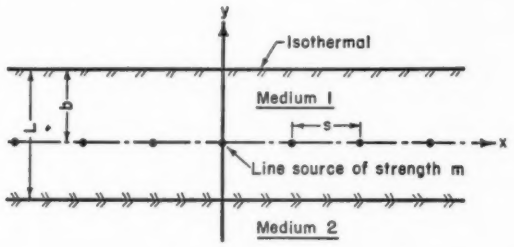


FIG. A-4. A ROW OF UNIFORMLY SPACED SOURCES IN A SLAB WITH ONE SURFACE ISOTHERMAL AND THE OTHER BOUNDED BY A SEMI-INFINITE MEDIUM

While Equation A-22 is a complete mathematical solution, the double summation presents a difficult problem of evaluation. Only for the special conditions of $\mu = 0$ and $\mu = \pm 1$ is comparative simplicity attained, as will be developed.

Problem 4. The Heat Flow Rates for the System of Problem 3 With the Upper Surface Isothermal, Fig. A-4

The rate of heat flow per unit area in the y direction at any point is

$$\frac{q_y}{A} = -k_1 \frac{\partial \theta}{\partial y}$$

Applying Equation A-18 to a single source and differentiating, there results

$$\left(\frac{q_y}{A}\right)_n = -\frac{m}{4\pi}\left[\frac{-2y}{(x-ns)^2+y^2} + \frac{2(y-2b)}{(x-ns)^2+(y-2b)^2} + \sum_{K=-\infty}^{K=+\infty}\left(\frac{2\mu^K(y-2b+2KL)}{(x-ns)^2+(y-2b+2KL)^2}\right.\right.$$

$$\left.\left. + \frac{2\mu^K(y-2b-2KL)}{(x-ns)^2+(y-2b-2KL)^2} - \frac{2\mu^K(y-2KL)}{(x-ns)^2+(y-2KL)^2} - \frac{2\mu^K(y+2KL)}{(x-ns)^2+(y+2KL)^2}\right)\right] \qquad (A\text{-}23)$$

TABLE A-2—POSITION AND STRENGTHS OF FIRST FEW IMAGES FOR IMAGES AT VARIOUS POSITIONS WITH THE PLANES $y = b$ AND $y = b - L$ BEING NON-ISOTHERMAL

Y	STRENGTH
0 (Source)	$+m$
$2b$	$-\mu_2 m$
$2L$	$+\mu_2\mu_1 m$
$2L + 2b$	$-\mu_2^2\mu_1 m$
$4L$	$+\mu_2^2\mu_1^2 m$
$4L + 2b$	$-\mu_2^3\mu_1^2 m$
$6L$	$+\mu_2^3\mu_1^3 m$
•	•
•	•
$2b - 2L$	$-\mu_1 m$
$-2L$	$+\mu_1\mu_2 m$
$2b - 4L$	$-\mu_1^2\mu_2 m$
$-4L$	$+\mu_1^2\mu_2^2 m$
$2b - 6L$	$-\mu_1^3\mu_2^2 m$
$-6L$	$+\mu_1^3\mu_2^3 m$

Primary interest is in the rate of heat flow through the upper (isothermal) slab surface. Setting $y = b$ and combining terms in Equation A-23 the rate of heat flow per unit area of surface at any point (x,b) is, for the n^{th} source alone,

$$\left(\frac{q}{A}\right)_{(x,b)_n} = \frac{m}{4\pi}\left[\frac{4b}{(x-ns)^2+b^2} + 4\sum_{K=1}^{K=+\infty}\left(-\frac{\mu^K(2KL-b)}{(x-ns)^2+(2KL-b)^2} + \frac{\mu^K(2KL+b)}{(x-ns)^2+(2KL+b)^2}\right)\right]$$

$$= \frac{m}{\pi}\left[\frac{b}{(x-ns)^2+b^2} + \sum_{K=1}^{K=+\infty}\mu^K\left(\frac{(2KL+b)}{(x-ns)^2+(2KL+b)^2} - \frac{(2KL-b)}{(x-ns)^2+(2KL-b)^2}\right)\right] \qquad (A\text{-}24)$$

To find the total heat flow rate for any individual source, Equation A-24 is integrated over half of a spacing interval, $x = 0$ to $x = s/2$. The heat flow will be the same for every half interval in the system of Fig. A-4. Integrating for unit length of source,

$$\frac{\pi}{m}\left(q_{\frac{s}{2}}\right)_n = \int_0^{\frac{s}{2}} \frac{b\,dx}{(x-ns)^2+b^2} + \int_0^{\frac{s}{2}} \sum_{K=1}^{K=+\infty}\mu^K\left(\frac{(2KL+b)}{(x-ns)^2+(2KL+b)^2}\right.$$

$$\left. - \frac{(2KL-b)}{(x-ns)^2+(2KL-b)^2}\right)dx \qquad (A\text{-}25)$$

This infinite series is uniformly convergent for $0 \leq x \leq s/2$ and hence the integration may be performed within the summation sign to yield

$$\frac{\pi}{m}\left(q_{\frac{s}{2}}\right)_n = \left[\tan^{-1}\left(\frac{x-ns}{b}\right)\right]_0^{\frac{s}{2}} + \sum_{K=1}^{K=+\infty}\mu^K\left[\tan^{-1}\left(\frac{x-ns}{2KL+b}\right) - \tan^{-1}\left(\frac{x-ns}{2KL-b}\right)\right]_0^{\frac{s}{2}}$$

$$= \tan^{-1}\frac{s(1-2n)}{2b} + \tan^{-1}\frac{ns}{b} + \sum_{K=1}^{K=+\infty}\mu^K\left(\tan^{-1}\frac{s(1-2n)}{2(2KL+b)} + \tan^{-1}\frac{ns}{(2KL+b)}\right)$$

$$-\sum_{K=1}^{K=+\infty}\mu^K\left(\tan^{-1}\frac{s(1-2n)}{2(2KL-b)} + \tan^{-1}\frac{ns}{(2KL-b)}\right) \qquad (A\text{-}26)$$

The surface heat flow rate for the half-spacing interval for all of the tubes or pipes is obtained by summing over all values of n. The area average rate of surface heat flow is obtained by dividing the total heat flow rate over the half-spacing interval by $s/2$. The result may be written

$$\left(\frac{q}{A_s}\right)_{av} = \frac{2m}{\pi s}\sum_{n=-\infty}^{n=+\infty}\frac{\pi}{m}\left(q_{\frac{s}{2}}\right)_n \qquad (A\text{-}27)$$

Problem 5. The Thermal Resistance Between a Pipe Surface and the Opposite Isothermal Slab Surface, for a Uniformly-Spaced Row of Pipes Located With Their Centers In or Near to the Slab Surface Which Is Common to an Adjacent Semi-Infinite Medium

For many analyses thermal resistance is a highly convenient quantity. This is determined as the ratio of the temperature potential Θ in Equation A-22 to the mean rate of surface heat flow Equation A-27.

$$R = \frac{\theta}{\left(\frac{q}{A_s}\right)_{av}} = \frac{\frac{m}{4\pi k_1}\left(\begin{matrix}\text{Double summation}\\ \text{of equation (A-22)}\end{matrix}\right)}{\frac{2m}{\pi s}\sum_{n=-\infty}^{n=+\infty}\frac{\pi}{m}\left(q_{\frac{s}{2}}\right)_n} = \frac{s}{8k_1}\frac{\left(\begin{matrix}\text{Double summation}\\ \text{of equation (A-22)}\end{matrix}\right)}{\sum_{n=-\infty}^{n=+\infty}\frac{\pi}{m}\left(q_{\frac{s}{2}}\right)_n} \qquad (A\text{-}28)$$

Equation A-28 is the general case.

For the particular condition of line sources in the lower slab surface, $b = L$, the mean thermal resistance for unit area of upper surface becomes

$$R = \frac{s}{\beta k_1} \left\{ \frac{\sum_{n=-\infty}^{n=+\infty} \left[\log_e \left[\frac{(x-ns)^2 + (y-2L)^2}{(x-ns)^2 + y^2} \right] + \sum_{K=1}^{K=+\infty} \mu^K \log_e \left[\frac{\left[(x-ns)^2 + (y-2L+2KL)^2\right]\left[(x-ns)^2 + (y+2KL)^2\right]}{\left[(x-ns)^2 + (y+2KL)^2\right]\left[(x-ns)^2 + (y-2KL)^2\right]} \right] \right]}{\sum_{n=-\infty}^{n=+\infty} \left[\tan^{-1} \frac{s(1-2n)}{2L} + \tan^{-1} \frac{ns}{L} + \sum_{K=1}^{K=+\infty} \mu^K \left(\tan^{-1} \frac{s(1-2n)}{2(2KL+L)} + \tan^{-1} \frac{ns}{(2KL+L)} \right) - \sum_{K=1}^{K=+\infty} \mu^K \left(\tan^{-1} \frac{s(1-2n)}{2(2KL-L)} + \tan^{-1} \frac{ns}{(2KL-L)} \right) \right]} \right\} \qquad (A\text{-}29)$$

The expression in its general form is very tedious to evaluate numerically. Two simplifying conditions appropriate for investigation as limiting cases are represented by $\mu = 0$ and $\mu = -1$. These are, respectively, the case of the earth and the slab having equal thermal conductivity and the case of the earth having zero conductivity. A third limiting condition, the case of the earth having infinite conductivity ($\mu = 1$) corresponds to an isothermal slab-earth interface and is a trivial case for the present problem.

When $\mu = 0$, Equation A-29 reduces to

$$R_{\mu=0} = \frac{s}{\beta k_1} \frac{\sum_{n=-\infty}^{n=+\infty} \log_e \left[\frac{(x-ns)^2 + (y-2L)^2}{(x-ns)^2 + y^2} \right]}{\sum_{n=-\infty}^{n=+\infty} \left[\tan^{-1} \frac{s(1-2n)}{2L} + \tan \frac{ns}{L} \right]} \qquad (A\text{-}30)$$

When $\mu = -1$, further manipulation produces simplification. The numerator series of Equation A-29 then may be broken up into four series:

(a) For K an even number, set $K = 2i$ and form two series:

$$+\sum_{n=-\infty}^{n=+\infty} \left[\log_e \left[(x-ns)^2 + (y-2L)^2 \right] + \sum_{i=1}^{i=+\infty} \log_e \left[(x-ns)^2 + (y+2(2i-1)L)^2 \right] \left[(x-ns)^2 + (y-2(2i+1)L)^2 \right] \right] \qquad (A\text{-}31)$$

$$-\sum_{n=-\infty}^{n=+\infty} \left[\log_e \left[(x-ns)^2 + y^2 \right] + \sum_{i=1}^{i=+\infty} \log_e \left[(x-ns)^2 + (y+4iL)^2 \right] \left[(x-ns)^2 + (y-4iL)^2 \right] \right] \qquad (A\text{-}32)$$

(b) For K an odd number, first set $K = 2i - 1$ and form the series:

$$-\sum_{n=-\infty}^{n=+\infty} \left[\sum_{i=1}^{i=+\infty} \log_e \left[(x-ns)^2 + (y+2(2i-2)L)^2 \right] \left[(x-ns)^2 + (y-4iL)^2 \right] \right] \qquad (A\text{-}33)$$

$$+\sum_{n=-\infty}^{n=+\infty} \left[\sum_{i=1}^{i=+\infty} \log_e \left[(x-ns)^2 + (y+2(2i-1)L)^2 \right] \left[(x-ns)^2 + (y-2(2i-1)L)^2 \right] \right] \qquad (A\text{-}34)$$

Expression A-33 then is written in two parts:

$$-\sum_{n=-\infty}^{n=+\infty} \left[\sum_{i=1}^{i=+\infty} \log_e \left[(x-ns)^2 + (y+2(2i-2)L)^2 \right] + \sum_{i=1}^{i=+\infty} \log_e \left[(x-ns)^2 + (y-4iL)^2 \right] \right]$$

Working with the first part only, make the substitution $i = j + 1$

$$\sum_{j=0}^{j=+\infty} \log_e\left[(x-ns)^2+(y+4jL)^2\right] = \log_e\left[(x-ns)^2+y^2\right] + \sum_{j=1}^{j=+\infty} \log_e\left[(x-ns)^2+(y+4jL)^2\right]$$

Substituting back in Equation A-33 there results:

$$-\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left[(x-ns)^2+y^2\right] + \sum_{j=1}^{j=+\infty} \log_e\left[(x-ns)^2+(y+4jL)^2\right] + \sum_{i=1}^{i=+\infty} \log_e\left[(x-ns)^2+(y-4iL)^2\right]\right]$$

Since there is no difference whether the summations above are made with respect to j or i, the final combination may be expressed as:

$$-\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left[(x-ns)^2+y^2\right] + \sum_{i=1}^{i=+\infty} \log_e\left[(x-ns)^2+(y+4iL)^2\right]\left[(x-ns)^2+(y-4iL)^2\right]\right] \quad \text{(A-33a)}$$

Expression A-33 is therefore the same as Expression A-32.

In a similar manner it can be established that Expression A-34 is the same as Expression A-31.

Expressions A-31 and A-34 have known summations. To recognize this make the substitutions:

In Equation A-29, set $(x - ns) = w$, $y - 2L = z$, $4L = a$

In Equation A-30, set $(x - ns) = w$, $y = z$, $4L = a$

The expressions then become:

$$+\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left[w^2+z^2\right] + \sum_{i=1}^{i=+\infty} \log_e\left[w^2+(z+ia)^2\right]\left[w^2+(z-ia)^2\right]\right] \quad \text{(A-31a)}$$

$$-\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left[w^2+z_1^2\right] + \sum_{i=1}^{i=+\infty} \log_e\left[w^2+(z_1+ia)^2\right]\left[w^2+(z_1-ia)^2\right]\right] \quad \text{(A-32a)}$$

Reference is here made to Equations A-9 and A-16 in the Appendix to the paper on Case I—Uniformly Spaced Pipes Buried Within a Solid Slab[1]. Series of the above form were shown therein to have summations as follows:

$$+\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left(\cosh\frac{2\pi w}{a} - \cos\frac{2\pi z}{a}\right) + \log_e\left(\frac{1}{\pi^2}\, a^2 a^4 2^4 a^4 3^4 \ldots\ldots\right)\right] \quad \text{(A-31b)}$$

$$-\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left(\cosh\frac{2\pi w}{a} - \cos\frac{2\pi z_1}{a}\right) + \log_e\left(\frac{1}{\pi^2}\, a^2 a^4 2^4 a^4 3^4 \ldots\ldots\right)\right] \quad \text{(A-32b)}$$

Since a is the same in A-31b and A-32b, addition of the two will cancel out the terms involving the products of a. Further, since A-32 and A-33 and correspondingly A-31 and A-34 are identical, the numerator series of Equation A-29 becomes:

$$\sum_{n=-\infty}^{n=+\infty}\left[2\log_e\left(\cosh\frac{\pi(x-ns)}{2L} - \cos\frac{\pi(y-2L)}{2L}\right) - 2\log_e\left(\cosh\frac{\pi(x-ns)}{2L} - \cos\frac{\pi y}{2L}\right)\right] \quad \text{(A-35)}$$

$$\text{But} \quad \cos\frac{\pi(y-2L)}{2L} = \cos\left(\frac{\pi y}{2L} - \pi\right) = -\cos\frac{\pi y}{2L}$$

---

[1] See Reference 1 of the paper.

NOMENCLATURE

b = distance between pipe axis and isothermal slab surface, in. or ft.

D = pipe outer diameter, in. or ft.

k_s = thermal conductivity of slab material, Btu per (hour) (square foot) (degree Fahrenheit per inch).

k_e = thermal conductivity of the adjacent medium (*e.g.*, the earth), Btu per (hour) (square foot) (degree Fahrenheit per inch).

K = integer.

L = slab thickness, in. or ft.

n = integer.

R = thermal resistance between pipe surface and opposite isothermal slab surface, defined in the text, Fahrenheit degree per Btu per (hour) (square foot).

s = pipe center-to-center spacing, in. or ft.

x,y = coordinate distances referred to some convenient pipe axis as the origin, in. or ft.

$\frac{q}{(A_s)_{av}}$ = area-mean rate of heat flow through the isothermal slab surface, Btu per (hour) (square foot).

θ_{p-s} = temperature difference between the outer pipe surface (on the side of the pipe facing the isothermal slab surface) and the opposite isothermal slab surface, degree Fahrenheit.

$$\mu = \frac{\frac{k_e}{k_s} - 1}{\frac{k_e}{k_s} + 1}$$

Hence, the summation A-35 becomes:

$$2\sum_{n=-\infty}^{n=+\infty}\left[\log_e\left(\cosh\frac{\pi(x-ns)}{2L} + \cos\frac{\pi y}{2L}\right) - \log_e\left(\cosh\frac{\pi(x-ns)}{2L} - \cos\frac{\pi y}{2L}\right)\right] \qquad (A\text{-}36)$$

For the condition of $\mu = -1$ all of the heat from the sources must flow out of the upper slab surface. This leads to the average heat flow rate

$$\left(\frac{q}{A_s}\right)_{av} = \frac{m}{s} \quad \ldots\ldots\ldots\ldots \quad (A\text{-}37)$$

Equation A-37 then replaces the denominator of Equation A-28 and the mean thermal resistance of the slab per unit of surface area is

$$R_{\mu=-1} = \frac{\frac{m}{4\pi k_t}}{\frac{m}{s}}\left[2\sum_{n=-\infty}^{n=+\infty}\log_e\left(\frac{\cosh\frac{\pi(x-ns)}{2L} + \cos\frac{\pi y}{2L}}{\cosh\frac{\pi(x-ns)}{2L} - \cos\frac{\pi y}{2L}}\right)\right] \qquad (A\text{-}38)$$

$$= \frac{s}{2\pi k_t}\sum_{n=-\infty}^{n=+\infty}\log_e\left(\frac{\cosh\frac{\pi(x-ns)}{2L} + \cos\frac{\pi y}{2L}}{\cosh\frac{\pi(x-ns)}{2L} - \cos\frac{\pi y}{2L}}\right) \qquad (A\text{-}39)$$

In evaluating the slab resistance the coordinates x and y appearing in the formulas are to be located in the slab on the outer surface of the pipe. For example, taking $x = 0$, $y = D/2$, where D = pipe outer diameter, places the point on the vertical diameter of the pipe.

DISCUSSION

MERL BAKER, Lexington, Ky. (WRITTEN): This report adds an important link in our knowledge of heat flow in panel heating and cooling embedment materials. The assumptions are judiciously made, and the favorable comparison between calculated results and analogue data is indeed gratifying.

Because of the importance of this paper I believe that it would be profitable for the authors to discuss in greater detail the following points:

1. *The formulation of the boundary conditions for the mathematical solutions.* Is the solution unduly complex for boundary conditions other than that of letting the thermal conductivity of the adjacent medium approach infinity?

2. *Interpretation of Fig. 6.* The assigned definition of R allows this term to be conveniently used in expressing pertinent relationships. What is the significance of ordinate values in Fig. 6 that are less than one?

3. *Influence of soil thermal conductivity on thermal resistance.* An explanation for the influence of k_s on R would be helpful in view of the assumption of zero heat flow to the ground. Except at the slab perimeter, the heat loss to the ground under the central portion of the slab does approach zero after the panel has been in operation for a period. Accordingly, the case taken has practical application.

In my opinion, this paper demonstrates the type of research needed in this field and continuance of the program should be encouraged.

N. S. BILLINGTON, London, England (WRITTEN): An attack upon the problem of floor temperatures in panel-heating systems has been made in Britain using an electrical analogue of the resistance network type. A full account of the work is to be published in *I.H.V.E. Journal** but a brief summary may be of interest.

The conditions used in the British work were somewhat similar to, though not identical with, those of the authors. The pipe temperature was constant at 100 F, and the air above the floor was held at 60 F. The surface of the floor was thus not at a uniform temperature. The conductivities of the concrete and earth were assumed to be the same, namely, 12.5 Btu/(hr) (sq ft) (F deg/in.). Pipe size and spacing, and the depth of *bury* were varied.

Fig. A shows the emission from the floor, with 1-in. o.d. pipes, expressed in Btu per (hour) (square foot) and per degree difference between pipe and air temperatures. The data from Table 1 of the paper have been indicated by crosses. There is seen to be a fair measure of agreement. In order to convert the British emissions into the resistance used by the authors, it is necessary to take the reciprocal and then subtract the surface resistance which was 0.6 (ft hr) (deg F) per Btu.

Fig. B shows how the surface temperature varies from point to point across the slab, and illustrates the extent of the departure from the isothermal condition postulated in the paper.

W. P. CHAPMAN, Pittsburgh, Pa. (WRITTEN): My only comment is that the *solid conducting medium adjacent* to the underside of the slab, should *not* be considered as earth. For some time now, pipe manufacturers have been stressing complete embedment of the pipe in the concrete (as shown in Fig. 2 of the paper). Construction as shown in Fig. 1 may lead to external corrosion, hence should be considered inferior practice.

If the paper is modified to state that the adjacent conduction medium may be a sub-slab on the bearing surface, *but not the fill or earth*, then the intent of the paper can be retained, and no improper construction practice will be described. We feel that, at the very least, some paragraph should be added explaining that the pipe should not be exposed to the earth.

---

* Floor-Panel Heating—Some Design Data, By N. S. Billington (*Journal of the Institution of Heating and Ventilating Engineers*, Vol. 21, No. 218, Oct. 1953, p. 256).

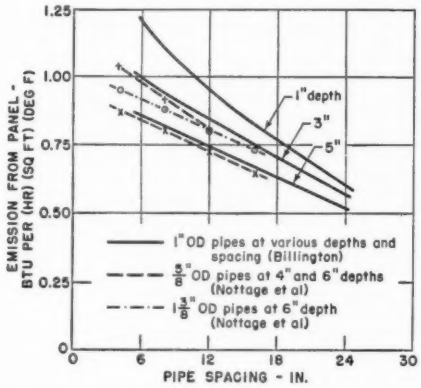


FIG. A. HEAT EMISSION FROM FLOOR PANELS

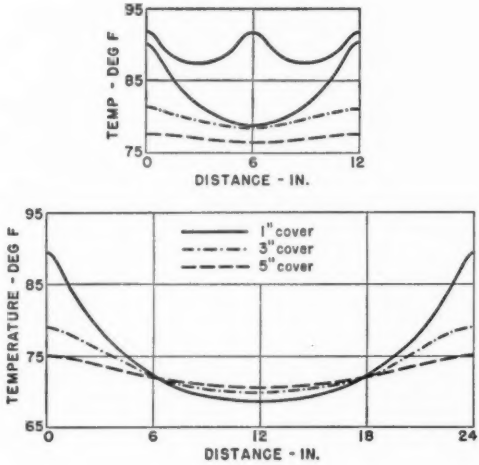


FIG. B. SURFACE TEMPERATURE DISTRIBUTION ON CONCRETE PANELS WITH 1-IN. O.D. PIPES AT VARIOUS CENTERS

Above: 6-in. and 12-in. centers
Below: 24-in. centers

AUTHORS' CLOSURE (H. B. Nottage): Mr. Chapman has raised a timely caution concerning construction practices. We certainly do not recommend placing pipes directly on the earth.

The present problem, Case II, and the previous analysis, Case I*, need to be completed by the projected Case III. Case III will cover pipes located near the interface but completely embedded in the concrete.

The medium below the floor slab can be considered as a sub-slab as long as its thickness is equal to or greater than the slab thickness, and provided that its thermal conductivity does not exceed the thermal conductivity of the slab itself. This suggestion is made for order-of-magnitude guidance.

In regard to the questions presented by Doctor Baker:

1. Reference should be made to the Appendix for a direct answer to the form of the mathematical solution.

2. In Fig. 6, ordinate magnitudes less than unity enter because L is measured to the pipe axis. When the pipe is very close to the surface, the thermal resistance of the two-dimensional conduction path is less than that of a unidimensional path of length L.

3. The influence of k_e upon R can be demonstrated by a detailed mapping of the isotherms and heat flow paths. As long as these enter the so-called earth, k_e will have an influence. Conduction paths which leave the pipe surface and enter the earth turn upward between the pipes and run to the upper slab surface. Again, details require study of the Appendix.

The need is recognized to study finite slabs on the earth. This is part of the long-range program initially mapped out and can be undertaken upon authorization.

The importance of transient and periodic conditions in practice must not be forgotten. These, also, can be studied as circumstances may dictate.

---

* See Reference 1 of the paper.

In Memoriam 1953

Name	Joined
Homer Addams *(Life Member, Presidential Member* and *Charter Member)*, Philadelphia, Pa.	1895
Albert E. Bauer, St. Paul, Minn.	1935
Charles A. Booth *(Life Member)*, Buffalo, N. Y.	1917
Roy Brauer, Pittsburgh, Pa.	1926
Cecil I. Cady, New York, N. Y.	1953
Clarence J. Carlson, Indianapolis, Ind.	1944
Harry Christenson, Battle Creek, Mich.	1931
Sherman F. Coghlan, Los Angeles, Calif.	1937
LeRoy E. Cox, New York, N. Y.	1944
Henry S. Crosby, Minneapolis, Minn.	1946
Frank C. Culver, Seattle, Wash.	1945
James J. Dannehy, Chicago, Ill.	1946
F. Norman Darby, Philadelphia, Pa.	1948
James H. Eagen, McKeesport, Pa.	1949
Frank J. Feely, Grosse Pointe, Mich.	1929
John B. Fehlig *(Life Member)*, Kansas City, Mo.	1918
Peter Fleury, Hamden, Conn.	1949
James J. Fraser, Toledo, Ohio	1936
Harry E. Gerrish *(Life Member)*, Minneapolis, Minn.	1910
Frederick E. Giesecke *(Life Member* and *Presidential Member)*, New Braunfels, Tex.	1913
George A. Gray, Ottawa, Ont., Canada	1924
William K. Griggs, Philadelphia, Pa.	1951
John P. Haffey, Los Angeles, Calif.	1947
Louis A. Harding *(Life Member* and *Presidential Member)*, Buffalo, N. Y.	1911
B. T. Hauer, Cincinnati, Ohio	1946
Daniel W. Hickey, St. Paul, Minn.	1931
Harry B. Joyce, Erie, Pa.	1944
Clarence F. Kessler, Ann Arbor, Mich.	1938
M. Felix Koster	1950
Gustus L. Larson *(Life Member* and *Presidential Member)*, Madison, Wis.	1923
Walter Leek *(Life Member)*, Vancouver, B. C., Canada	1903
Eugene J. Lefebvre, Westmount, Que., Canada	1937
Fred G. Little, Los Angeles, Calif.	1946
Eugene L. Long,* Columbus, Ohio	1944
George D. MacDonald, Holyoke, Mass.	1952

* Died in 1952 but Society not notified until 1953.

In Memoriam 1953

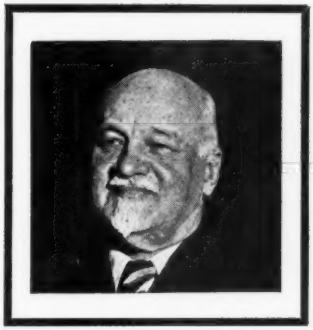
(continued)

NAME	JOINED
George R. Mansell, Toronto, Ont., Canada	1947
Charles E. Mayette, San Francisco, Calif.	1949
David E. McCulley, Omaha, Neb.	1941
Robert M. McDonald, Windsor, Ont., Canada	1952
William C. McNeill, Oklahoma City, Okla.	1951
Phillip E. Meidenbauer, Jr., Buffalo, N. Y.	1947
Clark T. Morse *(Life Member),* Detroit, Mich.	1913
Gail V. Murray, Monmouth, Ore.	1952
Floyd J. Parker, Jackson, Miss.	1953
Lyle G. Parrott *(Life Member),* Detroit, Mich.	1922
Frank H. Patterson, Grosse Pointe, Mich.	1942
R. W. Reed,* Jackson, Mich.	1949
Roland R. Reutling, Sr., Washington, D. C.	1946
John F. Reynolds,* Norfolk, Va.	1945
George B. Richmond, Memphis, Tenn.	1946
Ronald W. Rogers, Winnipeg, Man., Canada	1952
DeWitt C. Ruff* *(Life Member),* St. Paul, Minn.	1922
Ruben B. Saxon, Omaha, Neb.	1941
Edward H. Seelert, Minneapolis, Minn.	1935
Michael B. Shea *(Life Member),* Wayne, Pa.	1930
William K. Shine, Minneapolis, Minn.	1949
Everett T. Simonson, Dayton, Ohio	1950
C. Warner Smith, Philadelphia, Pa.	1948
Edward O. Smith, Pittsburgh, Pa.	1947
Franklyn Spiese, Seattle, Wash.	1952
Nichols L. Spinelli, Baltimore, Md.	1949
Ralph T. Squier,* Minneapolis, Minn.	1947
Edwin Sternberg, New York, N. Y.	1931
Evan E. Strath, Dearborn, Mich.	1950
Cyril Tasker, Cleveland, Ohio	1935
William R. Teller, Cleveland, Ohio	1946
Elmer Torok, Bristol, Tenn.	1936
Murray Waldman, Brooklyn, N. Y.	1950
Charles D. Waldon *(Life Member),* Toronto, Ont., Canada	1932
Olin A. Warren, Syracuse, N. Y.	1946
Louis A. Welch, Jr. *(Life Member),* Schenectady, N. Y.	1929
James Wilson, Montreal, Que., Canada	1942
Walter Yates *(Life Member),* Worsley, Lancashire, England	1902

* Died in 1952 but Society not notified until 1953.

Homer Addams

1873-1953



The unexpected news of the passing on July 3 of Dr. Homer Addams at the age of 80 greatly shocked the Officers, Council, and members of the Society. A Charter Member and Past President of A.S.H.V.E., Dr. Addams was one of the 75 organizers of the Society. At the time of his death he was chairman of the Board of Directors, Fitzgibbons Boiler Co., Inc., New York, N. Y.

Dr. Addams was born on June 22, 1873, in Addamstown, Pa., a town established by his ancestors in the early 1700's. He began his business career at the age of 15 after completion of preliminary education in public schools in the Addamstown area and in the Schuylkill Seminary. The desire for education was so great that he managed to combine business with study, taking whatever business and engineering courses he could in his off-hours.

His first job was as a helper, installing heating plants in Eastern Pennsylvania. In 1895, after completing his education and acquiring his basic experience, he joined the staff of Barber and Ross, a heating contracting firm of Washington, D. C., as an estimator and engineer in charge of construction. Dr. Addams joined a group of cast-iron boiler manufacturers in Philadelphia in 1896 and handled development, engineering and research of their products. During that same year, he was among that enterprising group of engineers who met in New York City to form THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, thus contributing a new concept of comfort, health and well-being for people everywhere.

In 1903, Dr. Addams was employed by the Kewanee Boiler Co. of Illinois and was sent to Washington as a special representative to contact Government agencies and to act as a consultant on various heating projects. He rejoined the Kewanee Co. in 1911 to set up a New York branch and also to serve as sales manager of the operation. From 1911-18, he spent much time in Washington as a liaison representative between the company and the Government on the design and development of boilers and hot water heaters.

Foreseeing the extended growth of construction along the eastern seaboard and the prospect of sizeable business volumes, Dr. Addams joined with the Kewanee Boiler Co. of Illinois in 1920 to form a subsidiary company known as

the Kewanee Boiler Co. of New York. His initial position with the new firm was vice president, but in 1921 he became president. The same year he acquired the Fitzgibbons Boiler Co., of Oswego, N. Y., and also became its president. He resigned as president in 1947 and was succeeded by his son, Paul K. Addams. Concurrently he became chairman of the Board of Directors of the firm and pursued his responsibilities with the same fervor that marked his long and outstanding career.

Dr. Addams' career has closely paralleled that of A.S.H.V.E., being identified with the advancement of engineering practice, research and development techniques in the field of heating and air conditioning. He was honored by the Society in 1952 when he received the F. Paul Anderson Medal in recognition of his notable contributions to the advancement of the art in the fields of boiler design and performance. In the previous year, Dr. Addams was awarded the Degree of Doctor of Science from Albright College, Reading, Pa.

Distinguished by an outstanding record of service to the Society, Dr. Addams participated in its work for many decades. He served as President in 1924; First Vice President in 1923; Treasurer 1915-22; and on the Council 1915-22. In addition, he was active in the founding of the Society's Research Laboratory at the United States Bureau of Mines, Pittsburgh, Pa., in 1919, and was a member of the first Guide Committee in 1922. He served the Society in many capacities and among his various committee assignments were the following:

1916-17	Executive Committee
	Committee on Education and Publicity
1918-19	Finance Committee
1918-25	Committee on Code for Testing Low Pressure Heating Boilers
1919-20	Executive Committee
1919-26	Committee on Research
1919-25	Committee to Confer with *A.S.M.E.* Boiler Code Committee on Heating Boilers
1920	Membership Committee
1921-23	Finance Committee
1922-23	Executive Committee
1922	Treasurer, Executive Committee on Research
	Treasurer, Guide Publication Committee
1925-53	Advisory Board
1926	Nominating Committee
1949	F. Paul Anderson Award Committee

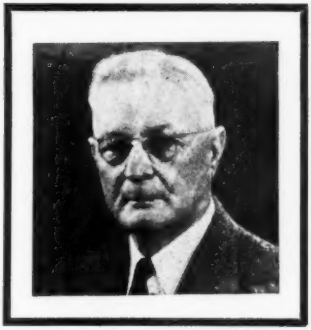
He also held membership in the *American Society of Mechanical Engineers,* the *American Uniform Boiler Law Society,* the *Steel Boiler Institute, Inc.,* the Engineers Club of New York, the Engineers Club of Philadelphia and the Greenleaf Lodge, F. & A. M., Allentown, Pa.

Dr. Addams will be keenly missed by his friends and associates in the Society to whom he was a constant source of inspiration. He was one of the Society's great leaders and his ideals will be cherished by his fellow members. Dr. Addams' name will go on record as one of the Society's immortals.

Surviving Dr. Addams are his wife, the former Elizabeth Swartz and his son Paul K. Addams.

Dr. Frederick E. Giesecke

1869-1953



The many Society friends and associates of Dr. Frederick E. Giesecke of New Braunfels, Tex., were greatly saddened to learn of his death on June 27, at the age of 85. The "Pal", as he was fondly called because of his outstanding benevolence to all those he came in contact with, was an eminent authority on heating and ventilating, being distinguished for his researches in the fields of hot water heating, heat transmission and air conditioning for comfort. He was also considered an expert on radiant heating methods.

A Past President and Life Member of A.S.H.V.E., Dr. Giesecke was born on January 28, 1869, in Washington County, Tex., and received his preparatory education in the New Braunfels and San Antonio schools. He began his professional career at the A. & M. College of Texas, College Station, Tex., from which he received his M.E. in 1890. In 1904 he received an S.B. degree from the Massachusetts Institute of Technology, and from 1886 to 1912 he was a teacher at the A. & M. College of Texas. In 1906-7 he studied European heating and ventilating methods in Germany. As college architect he designed and directed installations of heating systems in new buildings from 1908 to 1912.

Dr. Giesecke, in 1913, became professor of Architecture at the University of Texas, Austin, as well as consulting architect. He received a doctor's degree from the University of Illinois in 1924. From 1927 to 1939, he was Director of the Engineering Station at A. & M. College of Texas. At the time of his death, Dr. Giesecke was Professor Emeritus at the College and a consulting engineer.

In 1942 Dr. Giesecke received the F. Paul Anderson Gold Medal awarded by the Society for distinguished scientific achievement. The Committee on Award honored Dr. Giesecke for his notable contributions to the advancement of heating based on his research work in the fields of heat transfer and hot water heating, and for his services to the Society as a member and officer.

He was the author of several technical books, including *The Design of Gravity-Circulation Water Heating Systems,* published in 1926, and the joint author with A. Mitchell and H. G. Spencer of *Technical Drawing,* and *Technical Drawing Problems,* and other technical books and papers.

Dr. Giesecke was President of the Society in 1940, First Vice President in 1939, Second Vice President in 1938, served on the Council from 1932-41 and on the Advisory Board from 1941-53. He also served as Chairman of the Executive Committee and on many Research Technical Advisory Committees of the Society. He was secretary of the Austin, Tex., Chapter from 1917-1919. Extremely active in Society affairs, Dr. Giesecke served on numerous committees as follows:

1921-22	Sub-Committee on Pipe Sizes
1922-24	Committee on Steam and Return Main Sizes
1925	Committee on Pipe Sizes
1929-31	Committee on Research
1929-30	T.A.C. on Pipe Sizes for Heating Systems
1931-34	T.A.C. on Pipe and Tubing (Sizes) Carrying Low Pressure Steam or Hot Water
1931	T.A.C. on Radiation—Direct and Indirect
1932-34	Committee on Research
1932	T.A.C. on Infiltration in Buildings
1933	Guide Publication Committee
1934	Executive Committee
1934-36	Cooperating Committee-A.S.H.V.E. Representative on National Research Council
1935-36	Committee of the Council on Meetings
1936	T.A.C. on Comfort Requirements for Summer Cooling
1937-41	Executive Committee of the Council
1937	T.A.C. on Comfort Air Conditioning-OH-22 T.A.C. on Intermittent Heating-1P-20
1939, 41, 1943, 45-47, 1949	F. Paul Anderson Award
1939	Committee on Engineering Scholarships
1943	T.A.C. on Radiation with Gravity Air Circulation
1943-45	T.A.C. on Flow of Fluids through Pipes and Fittings
1944-46	Committee on Research
1944	T.A.C. on Heat Requirements of Buildings
1944-46	T.A.C. on Radiation and Comfort
1947-53	T.A.C. on Panel Heating and Cooling-Group A-Heat Distribution Within and Behind the Panel

Listed in *Who's Who In Engineering*, Dr. Giesecke was held in high esteem by leaders in both the engineering and architectural fields. He was a member of *A.S.E.E., A.A.A.S., A.S.C.E., A.S.M.E.* and the honorary fraternities, *Sigma Xi* and *Tau Beta Pi*.

Dr. Giesecke will be remembered not only for his many fine contributions as an educator, engineer and architect, but for his benevolence to all those who came in contact with him. He was always available for counsel and help. The Officers and Council and members of the Society will deeply feel the loss of Dr. Giesecke and will long cherish his memory.

Surviving Dr. Giesecke are his wife, the former Hulda C. Gruene, and three daughters.

Louis A. Harding

1876-1953

It was with deep sorrow that the Society learned of the death of Louis Allen Harding at his home in Buffalo, N. Y., on July 8 at the age of 76. He joined the Society in 1911, served as its President in 1930, and was a Life Member. Mr. Harding was born on October 16, 1876, in Factoryville, Pa. He received his preliminary education at the Hackettstown Seminary and later attended Penn State College, State College, Pa., where in 1899 he received his degrees of B.S. and M.E. For several years he was a draftsman for Dickson Mfg. Co., in 1901 joined Lackawanna Steel Co., and later was superintendent of Washeries for Lackawanna Coal & Coke Co. Between engineering engagements he was an instructor at Cornell University in 1902, professor of machine design and head of the mechanical engineering department at Penn State College 1905-06, and professor of Mechanical Engineering at the University of Illinois 1913-17. He was a member of the firm of Pennsylvania Engineering Co., and later chief engineer of Armstrong Cork Co.

Mr. Harding came to Buffalo and entered the contracting field as a partner of Cummings and Harding, then became chief engineer of John Cowper Co. In 1921 the consulting firm of Harding & Crea was organized, and in 1925 he organized the L. A. Harding Construction Co. of which he was president. He was president of Harding-Carlton Inc., an officer of the Wilkinson-Harding Corp., a real estate firm, and the Leach Steel Corp., of Rochester, a structural firm. In his consulting and construction work he was responsible for the design of the mechanical equipment for hundreds of plants, and built many industrial buildings.

A man of great versatility as an engineer, scientist, educator and scholar, Mr. Harding was acclaimed for his leadership and his organizing ability. He was a prodigious reader and a frequent contributor to engineering and scientific periodicals. As co-author with Dr. A. C. Willard he wrote two volumes on *Mechanical Equipment of Buildings, Steam Power Plants,* and more recently books on mathematics and navigation. He contributed sections to *Kent's Engineers Handbook* and *Kidder's Architects' and Builders' Handbook.*

In 1937 he became interested in local politics and although a life long Republican he was a key member of Democratic Mayor Thomas Hollings' administration, serving for nearly four years as Commissioner of Public Works of the City of Buffalo. He brought in a new era where ability, economy and integrity were the yardsticks of his accomplishments which are legendary.

Mr. Harding largely designed and checked plans for the Airport Administration Building. The Memorial Auditorium was built under his personal inspection.

Following his community service he returned to his engineering business and resumed his writing, but had been in retirement for the past few years.

Mr. Harding became a member of the Society in 1911 and was always active in its affairs. He was First Vice President in 1929; Second Vice President in 1928; and served on the Council from 1922-31. Mr. Harding was one of the founders of Western New York Chapter and served as president in 1921, and on its Board of Governors from 1927-35.

During his Society membership of 42 years he served on many committees as a member or as chairman, among which are the following:

1912-16	Committee on Tests
1917	Committee on Code for Testing Low Pressure Heating Boilers
1919	Nominating Committee
1919-22	Committee to Confer with *A.S.M.E.* Boiler Code Committee on Heating Boilers
1920-21	Committee on Steam and Return Sizes
1921-22	Committee on Standard Code for Testing Heating Systems
1923-28	Committee on Code of Minimum Requirements for Heating and Ventilating of Buildings
1923	Committee on Furnace Heating
1924-28	Chairman, Publication Committee
1925-26	Guide Publication Committee
1926-27	Committee on Heat Transmission Through Buildings
1928-31-32	Committee on Heat Transmission
1929-33	Executive Committee of the Council
1929-32	
33-34	Committee on Research
1929-30-34	Committee for Interpreting Code for Rating Low Pressure Heating Boilers
1931-32	Committee on Meetings
1931	Committee on Definitions
1932-52	Advisory Council
1932	Committee on Ventilation of Bus Terminals
1932-34	Committee on Ventilation Standards
1933	Committee on Nomenclature
1935	F. Paul Anderson Award Committee
1936-37	T.A.C. on Insulation
1936-37	Chairman A.S.H.V.E.-*A.S.R.E.* Committee on Standards for Air Conditioning Applications

His unselfish service to the Society will be treasured by his many friends and associates and his name will be indelibly inscribed in the records of A.S.H.V.E.

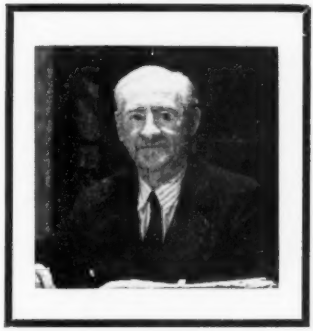
He was a prominent Mason, a fellow of *The American Society of Mechanical Engineers,* a member of the *American Society of Civil Engineers,* Sigma Psi, Phi Kappa Sigma, Pi Tau Sigma, as well as a number of other societies and clubs. He is listed in *Who's Who in America* and *Who's Who in Engineering.*

Those who had the opportunity of working with Mr. Harding know the inspiration of his leadership, his willingness to share his vast knowledge and experience with others, his soft spoken manner, but as a hard fighter for the principles he believed in. His friendly presence and delightful wit will be greatly missed by his friends who are legion, and regard him as the complete engineer.

He is survived by his wife, Mrs. Charlotte Hanes Harding.

Gustus L. Larson

1881-1953



The news of the unexpected death Sunday, August 16, of Professor Gustus L. Larson, widely known educator, author and consulting engineer, was received with deep regret by the Society. A past president of A.S.H.V.E., Professor Larson was a highly respected authority on heating and air conditioning, his teachings and research studies influencing many. He was 72 years old at the time of his death.

Born in Sweden, Professor Larson came to the United States at the age of nine and was naturalized at 21. He was graduated from the University of Idaho in 1907 with a B.S. in electrical engineering, and was employed by the General Electric Co., Schenectady, N. Y., as a test engineer from 1907 to 1909. In 1909 he returned to Idaho to accept an appointment to teach engineering at the University, and began his long and successful career in the educational field.

As a result of an extensive research investigation of recirculation of washed air in an enclosure, Professor Larson was granted a mechanical engineering degree by the University of Wisconsin in 1915. From 1915 to 1920 he was associate professor of steam and gas engineering at the University, and in 1920 was appointed a full professor and chairman of the Department of Mechancial Engineering.

As a consultant engineer, Professor Larson was instrumental in designing mechanical equipment for many outstanding private and public buildings in Wisconsin and Illinois. Among these were the Wisconsin Power and Light Building, Tenney Office Building, Memorial Union Building, University of Wisconsin Field House, and Manchester's Department Store, all in Madison; Dane County Sanitarium; Central Illinois Public Service Building, Springfield, Ill.; and numerous public schools in the area. He also designed the mechanical equipment of several large hospital units built in Milwaukee, Madison and Janesville.

In 1925 Professor Larson was appointed by the Wisconsin Industrial Commission to assist in drafting Wisconsin's heating and ventilating code, and in 1936 helped to formulate its air conditioning code. During Franklin D. Roose-

velt's administration he was called to Washington to serve on the Conference on Home Building and Ownership.

Professor Larson's many contributions to his profession were highlighted in May of 1951, when more than 200 persons gathered in Madison to honor him upon his retirement as Professor of Mechanical Engineering at the University of Wisconsin. Among those who paid tribute at this time were students, former students, faculty colleagues, Society friends, and others who had benefited by knowing Professor Larson. A few weeks earlier, the Wisconsin Chapter of A.S.H.V.E. had established the Gustus L. Larson Award, to be presented annually to an outstanding University of Wisconsin engineering student. Professor Larson had served for many years on the Board of Governors of the chapter.

Among other engineering societies in which Professor Larson held active membership were: *American Society of Mechanical Engineers, Engineering Society of Wisconsin* of which he was president in 1933, *Society for the Promotion of Engineering Education, American Association of University Professors* and the *National Association of Power Engineers.*

He was also a member of the following honorary and social fraternities: Phi Delta Theta, Triangle, Tau Beta Pi, Sigma Xi and Phi Kappa Phi. From 1926 to 1929 he acted as national president of Pi Tau Sigma, national honorary fraternity in mechanical engineering.

His service to the Society, extending over many years, began in 1928 and was climaxed in 1936 when he was elected president of A.S.H.V.E.. From 1929 to 1936 he was a member of the Council. In 1934 he was elected Second Vice President of the Society and in 1935, First Vice President. Among the many committees on which he served actively were the following:

1928	T.A.C. on Infiltration of Air into Buildings (Chairman)
1929	Council, Committee on Membership
	T.A.C. on Infiltration (Chairman)
1930	Council, Publication Committee (Chairman)
	T.A.C. on Infiltration (Chairman)
	Nominating Committee (Chairman)
1931	Council, Committee on Meetings Programs (Chairman)
	T.A.C. on Infiltration (Chairman)
1932	Council, Chairman of Committee on Research
1933	Council, Chairman of Committee on Research
	Council, Executive Committee
1934	Second Vice President
	Guide Publication Committee
	Committee on Research
	Research Finance Committee
1935	First Vice President
	Guide Publication Committee (Chairman)
	F. Paul Anderson Award Committee (Chairman)
1937	Advisory Council (Chairman)
1937-38	F. Paul Anderson Award Committee
1939	Committee on Engineering Scholarships

INDEX

A

Accountant's Report 4
ACHENBACH, P. R., Discussions on: Heat Exchanges in a Ceiling Panel Heated Room 219
Heat Transmitted to the I=B=R Research Home from the Inside Chimney..... 110
ADDAMS, HOMER, In Memoriam (1873-1953)..... 549
Air Conditioning, Room Air Distribution Research for Year 'Round, Part I, by S. F. GILMAN, H. E. STRAUB, A. E. HERSHEY and R. B. ENGDAHL..... 151
Air Distribution Research for Year 'Round Air Conditioning, Room, Part I, by S. F. GILMAN, H. E. STRAUB, A. E. HERSHEY and R. B. ENGDAHL..... 151
Air Velocities in Ventilating Jets, by G. L. TUVE..... 261
Discussion 282
ALGREN, A. B., Discussion on Heat Exchanges in a Floor Panel Heated Room.... 508
ALGREN, A. B., E. F. SNYDER, JR., and J. S. LOCKE, Field Studies of Floor Panel Control Systems 173
Aluminum Ducts, Friction Losses in Round Aluminum, by F. W. HUTCHINSON.... 127
AMBROSE, E. R., Discussion on Electrical Analogger Application to Heat Pump Process 369
Analogger Application to the Heat Pump Process, Electrical, by C. F. KAYAN.... 361
ANDERSON, D. B., Discussion on Thermal Performance of Frame Walls..... 461
Annual Meeting Proceedings..... 1
Annual Report of Committee on Research—1952..... 18
ANTONIO, J. L., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence..... 492
Automatic Permeance Measurement by the Permeometer, by F. A. JOY and A. W. SHERDON..... 435
Discussion 447
AVERY, L. T., Discussion on The Bacteria and Odor Control Problem in Occupied Spaces 93

B

Bacteria and Odor Control Problem in Occupied Spaces, The, by R. L. KUEHNER 77
Discussion 88
BAHNFLETH, D. R., H. T. GILKEY and R. W. ROOSE, Cooling a Small Residence With a Two-Horsepower Mechanical Condensing Unit..... 283
BAKER, MERL, Design and Performance of a Residential Earth Heat Pump..... 371
BAKER, MERL, Discussion on Heat Flow Analysis in Panel Heating or Cooling Sections 546
BERRY, N. E., Discussion on The Bacteria and Odor Control Problem in Occupied Spaces 90
BILLINGTON, N. S., Discussion on Heat Flow Analysis in Panel Heating or Cooling Sections 546
BROWN, H. G., H. G. CONN and W. G. COLBORNE, Pressure Losses in 4-inch Diameter Galvanized Metal Duct and Fittings..... 139
BURCH, L. A., Discussion on Further Studies of the Thermal Characteristics of Plaster Panels 523
By-Laws, Amendments to..... 306

C

Calculating the Friction in Pipe Lines, Two Methods of, by F. E. GIESECKE.... 49
CAMPBELL, E. K., Discussion on Heat Transmitted to the I=B=R Research Home from the Inside Chimney..... 111
CARLSON, G. F., and H. A. LOCKHART, Compression Tank Selection for Hot Water Heating Systems 55

CARROLL, J. R., Discussion on Heat Exchanges in a Ceiling Panel Heated Room 216
Ceiling Panel Heated Room, Heat Exchanges in a, by L. F. SCHUTRUM, G. V. PARMELEE and C. M. HUMPHREYS 197
CHANG, S. C., and F. C. HOOPER, Development of the Thermal Conductivity Probe 463
CHAPMAN, W. P., Discussions on: Heat Exchanges in a Ceiling Panel Heated Room 215
Heat Flow Analysis in Panel Heating or Cooling Sections 546
CHEWNING, R. C., Discussions on: Compression Tank Selection for Hot Water Heating Systems 74
Design and Performance of a Residential Earth Heat Pump 392
Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems 359
CHILDS, M. E., H. T. GILKEY and R. W. ROOSE, Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 473
Chimney, Heat Transmitted to the I=B=R Research Home from the Inside, by W. S. HARRIS and R. J. MARTIN 97
CHRISTESEN, M. C., Discussion on Compression Tank Selection for Hot Water Heating Systems 75
COLBORNE, W. G., H. G. CONN and W. G. BROWN, Pressure Losses in 4-inch Diameter Galvanized Metal Duct and Fittings 139
Committee on Arrangements, Fifty-ninth Annual Meeting 34
Semi-Annual Meeting, 1953 310
Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment, A, by TOHRU INOUYE, F. K. HICK, M.D., R. W. KEETON, M.D., J. LOSCH, and NATHANIEL GLICKMAN 35
Compression Tank Selection for Hot Water Heating Systems, by H. A. LOCKHART and G. F. CARLSON 55
Discussion 74
Condensing Unit, a Two-Horsepower Mechanical, Cooling a Small Residence with, by H. T. GILKEY, D. R. BAHNFLETH and R. W. ROOSE 283
Conductivity Probe, Thermal, Development of the, by F. C. HOOPER and S. C. CHANG 463
CONN, H. G., W. G. COLBORNE and W. G. BROWN, Pressure Losses in 4-inch Diameter Galvanized Metal Ducts and Fittings 139
CONOVER, E. W., Discussion on Design Data for Slat-Type Sun Shades for Use in Load Estimating 432
Control Systems, Floor Panel, Field Studies of, by A. B. ALGREN, E. F. SNYDER, JR., and J. S. LOCKE 173
Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit, by H. T. GILKEY, D. R. BAHNFLETH and R. W. ROOSE 283
Discussion 303
Cooling Tower, Performance of a Forced Draft, by B. H. SPURLOCK, JR. 311
CRANK, R. E., LINN HELANDER and SHEE-MANG YEN, Maximum Downward Travel of Heated Jets From Standard Long Radius *A.S.M.E.* Nozzles 241

D

DANIELSON, W. A., Discussions on: Automatic Permeance Measurement by the Permeometer 448
Design Data for Slat-Type Sun Shades for Use in Load Estimating 433
Heat Exchanges in a Floor Panel Heated Room 509
Performance of a Forced-Draft Cooling Tower 327
Thermal Performance of Frame Walls 460
Design and Performance of a Residential Earth Heat Pump, by MERL BAKER 371
Discussion 391
Design Data for Slat-Type Sun Shades for Use in Load Estimating, by G. V. PARMELEE and D. J. VILD 403
Discussion 432
Development of the Thermal Conductivity Probe, by F. C. HOOPER and S. C. CHANG 463
Discussion 472

DILL, R. S., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 491
Duct and Fittings, Galvanized Metal, Pressure Losses in 4-inch Diameter, by H. G. CONN, W. G. COLBORNE and W. G. BROWN 139
Ducts, Friction Losses in Round Aluminum, by F. W. HUTCHINSON 127

E

Earth Heat Pump, Residential, Design and Performance of a, by MERL BAKER 371
Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F, by TOHRU INOUYE, F. K. HICK, M.D., S. E. TELSER, M.D., and R. W. KEETON, M.D. 329
Discussion 342
EISENSTADT, RAYMOND, and W. A. HADLEY, Moisture Movement in Soils Due to Temperature Difference 395
Electrical Analogger Application to the Heat Pump Process, by C. F. KAYAN 361
Discussion 369
ENGDAHL, R. B., S. F. GILMAN, H. E. STRAUB and A. E. HERSHEY, Room Air Distribution Research for Year 'Round Air Conditioning 151
Engineering Research, Participation by the Colleges in, by E. A. WALKER 17
Environment, A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in, by TOHRU INOUYE, F. K. HICK, M.D., R. W. KEETON, M.D., J. LOSCH and NATHANIEL GLICKMAN 35
Environments of 80, 76, and 72 F, Effect of Relative Humidity on Heat Loss of Men Exposed to, by TOHRU INOUYE, F. K. HICK, M.D., S. E. TELSER, M.D., and R. W. KEETON, M.D. 329
Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems, by W. L. ROGERS 347
Discussion 357

F

FAHNESTOCK, M. K., Discussions on: Design and Performance of a Residential Earth Heat Pump 392
Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 342
FAUST, F. H., Discussion on Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 344
Field Studies of Floor Panel Control Systems, by A. B. ALGREN, E. F. SNYDER, JR., and J. S. LOCKE 173
Fifty-Ninth Annual Meeting 1
Accountant's Report 4
Council, Report of 1
Research Committee, 1952 Report of 18
Symposium on Research Administration 16
Treasurer's Report 3
Floor Panel Control Systems, Field Studies of, by A. B. ALGREN, E. F. SNYDER, JR., and J. S. LOCKE 173
Floor Panel Heated Room, Heat Exchanges in a, by L. F. SCHUTRUM, G. V. PARMELEE and C. M. HUMPHREYS 495
Forced-Draft Cooling Tower, Performance of a, by B. H. SPURLOCK, JR. 311
Frame Walls, Thermal Performance of, by G. O. HANDEGORD and N. B. HUTCHEON 449
FRANKS, C. V., H. B. NOTTAGE, L. E. HULBERT and L. F. SCHUTRUM, Heat Flow Analysis in Panel Heating or Cooling Sections 527
Friction Losses in Round Aluminum Ducts, by F. W. HUTCHINSON 127
FRIEND, W. F., Discussions on: Design and Performance of a Residential Earth Heat Pump 392
Electrical Analogger Application to Heat Pump Process 369
Performance of a Forced-Draft Cooling Tower 327

Fuel Consumption Analysis for Multi-Family Housing Projects, by RUTCHER SKAGERBERG and J. E. PHIFER 113
Discussion 125
Further Studies of the Thermal Characteristics of Plaster Panels, by L. F. SCHUTRUM and C. M. HUMPHREYS 511
Discussion 521

G

Galvanized Metal Duct and Fittings, Pressure Losses in 4-inch Diameter, by H. G. CONN, W. G. COLBORNE and W. G. BROWN 139
GAMBLE, C. B., Discussion on Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 345
GIESECKE, F. E., Discussion on Heat Exchanges in a Ceiling Panel Heated Room 218
GIESECKE, F. E., In Memoriam (1869-1953) 555
GIESECKE, F. E., Two Methods of Calculating the Friction in Pipe Lines 49
GILKEY, H. T., D. R. BAHNFLETH and R. W. ROOSE, Cooling a Small Residence With a Two-Horsepower Mechanical Condensing Unit 283
GILKEY, H. T., R. W. ROOSE and M. E. CHILDS, Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 473
GILMAN, S. F., H. E. STRAUB, A. E. HERSHEY and R. B. ENGDAHL, Room Air Distribution Research for Year 'Round Air Conditioning 151
GLICKMAN, NATHANIEL, TOHRU INOUYE, F. K. HICK, M.D., R. W. KEETON, M.D., and J. LOSCH, A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment 35
GORDON, P. B., Discussions on: Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 345
Further Studies of the Thermal Characteristics of Plaster Panels 523
Heat Exchanges in a Ceiling Panel Heated Room 219
Heat Exchanges in a Floor Panel Heated Room 507
Government Sponsored Research—A Canadian View, by R. F. LEGGET 17
GRANT, W. A., Discussion on Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 344
GREILING, W. W., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 492

H

HADLEY, W. A., and RAYMOND EISENSTADT, Moisture Movement in Soils Due to Temperature Difference 395
HANDEGORD, G. O., and N. B. HUTCHEON, Thermal Performance of Frame Walls 449
HARDING, L. A., In Memoriam (1876-1953) 553
Heat Exchanges in a Ceiling Panel Heated Room, by L. F. SCHUTRUM, G. V. PARMELEE and C. M. HUMPHREYS 197
Discussion 215
Heat Exchanges in a Floor Panel Heated Room, by L. F. SCHUTRUM, G. V. PARMELEE and C. M. HUMPHREYS 495
Discussion 507
Heat Flow Analysis in Panel Heating or Cooling Sections, Case II—Floor Slab on Earth with Uniformly Spaced Pipes or Tubes at the Slab-Earth Interface, by H. B. NOTTAGE, C. V. FRANKS, L. E. HULBERT and L. F. SCHUTRUM 527
Discussion 546
Heat Loss of Men Exposed to Environments of 80, 76, and 72 F, Effect of Relative Humidity on, by TOHRU INOUYE, F. K. HICK, M.D., S. E. TELSER, M.D., and R. W. KEETON, M.D. 329
Heat Pump Process, Electrical Analogger Application to, by C. F. KAYAN 361
Heat Pump, Residential Earth, Design and Performance of a, by MERL BAKER 371
Heat Transmitted to the I=B=R Research Home from the Inside Chimney, by W. S. HARRIS and R. J. MARTIN 97
Discussion 110

Heated Jets from Standard Long Radius *A.S.M.E.* Nozzles, Maximum Downward Travel of, by LINN HELANDER, SHEE-MANG YEN and R. E. CRANK.... 241
Heating Systems, Compression Tank Selection for Hot Water, by H. A. LOCKHART and G. F. CARLSON.... 55
HELANDER, LINN, SHEE-MANG YEN and R. E. CRANK, Maximum Downward Travel of Heated Jets from Standard Long Radius *A.S.M.E.* Nozzles.... 241
HERRINGTON, L. P., Discussion on Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F.... 342
HERSHEY, A. E., S. F. GILMAN, H. E. STRAUB and R. B. ENGDAHL, Room Air Distribution Research for Year 'Round Air Conditioning.... 151
HICK, F. K., M.D., TOHRU INOUYE, R. W. KEETON, M.D., J. LOSCH and NATHANIEL GLICKMAN, A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment.... 35
HICK, F. K., M.D., TOHRU INOUYE, S. E. TELSER, M.D., and R. W. KEETON, M.D., Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F.... 329
HOOPER, F. C., and S. C. CHANG, Development of the Thermal Conductivity Probe.... 463
Hot Water Heating Systems, Compression Tank Selection for, by H. A. LOCKHART and G. F. CARLSON.... 55
Housing Projects, Fuel Consumption Analysis for Multi-Family, by RUTCHER SKAGERBERG and J. E. PHIFER.... 113
HUEBNER, W. O., Discussion on Thermal Performance of Frame Walls.... 461
HULBERT, L. E., H. B. NOTTAGE, C. V. FRANKS and L. F. SCHUTRUM, Heat Flow Analysis in Panel Heating or Cooling Sections.... 527
HUMMEL, G. W., Discussions on: Design and Performance of a Residential Earth Heat Pump.... 392
Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F.... 345
Performance of a Forced-Draft Cooling Tower.... 327
HUMPHREYS, C. M., and L. F. SCHUTRUM, Further Studies of the Thermal Characteristics of Plaster Panels.... 511
HUMPHREYS, C. M., L. F. SCHUTRUM and G. V. PARMELEE, Heat Exchanges in a Ceiling Panel Heated Room.... 197
HUMPHREYS, C. M., L. F. SCHUTRUM and G. V. PARMELEE, Heat Exchanges in a Floor Panel Heated Room.... 495
HUTCHEON, N. B., and G. O. HANDEGORD, Thermal Performance of Frame Walls 449
HUTCHEON, N. B., Discussion on Moisture Movement in Soils Due to Temperature Difference.... 401
HUTCHINSON, F. W., Friction Losses in Round Aluminum Ducts.... 127

I

I=B=R Research Home, Heat Transmitted to the, from the Inside Chimney, by W. S. HARRIS and R. J. MARTIN.... 97
Industrial Research in Control of Environment, by W. H. KLIEVER.... 17
INGERSOLL, L. R., Discussion on Development of the Thermal Conductivity Probe 472
INOUYE, TOHRU, F. K. HICK, M.D., R. W. KEETON, M.D., J. LOSCH and NATHANIEL GLICKMAN, A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment.... 35
INOUYE, TOHRU, F. K. HICK, M.D., S. E. TELSER, M.D., and R. W. KEETON, M.D., Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F.... 329

J

JAMES, J. W., Discussion on Compression Tank Selection for Hot Water Heating Systems.... 75
Jets, Maximum Downward Travel of Heated, from Standard Long Radius *A.S.M.E.* Nozzles, by LINN HELANDER, SHEE-MANG YEN and R. E. CRANK 241
Jets, Ventilating, Air Velocities in, by G. L. TUVE.... 261

JONES, A. T., Discussion on Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems 357
JORN, A. T., Discussion on Heat Exchanges in a Ceiling Panel Heated Room 219
JOY, F. A., and A. W. SHERDON, Automatic Permeance Measurement by the Permeometer 435

K

KAYAN, C. F., Discussion on Further Studies on the Thermal Characteristics of Plaster Panels 521
KAYAN, C. F., Electrical Analogger Application to the Heat Pump Process 361
KEETON, R. W., M.D., TOHRU INOUYE, F. K. HICK, M.D., and S. E. TELSER, M.D., Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 329
KEETON, R. W., M.D., TOHRU INOUYE, F. K. HICK, M.D., J. LOSCH and NATHANIEL GLICKMAN, A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment 35
KERN, R. T., Discussion on Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems 358
KLIEVER, W. H., Industrial Research in the Control of Environment 17
KROEKER, J. D., Discussions on: Design and Performance of a Residential Earth Heat Pump 391
Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 344
KUEHNER, R. L., The Bacteria and Odor Control Problem in Occupied Spaces 77

L

LARSON, G. L., In Memoriam (1881-1953) 557
LEGGET, R. F., Government Sponsored Research—A Canadian View 17
Load Estimating, Design Data for Slat-Type Sun Shades for Use in, by G. V. PARMELEE and D. J. VILD 403
LOCKE, J. S., A. B. ALGREN and E. F. SNYDER, JR., Field Studies of Floor Panel Control Systems 173
LOCKHART, H. A., and G. F. CARLSON, Compression Tank Selection for Hot Water Heating Systems 55
LOCKLIN, D. W., Discussion on Pressure Losses in 4-inch Diameter Galvanized Metal Duct and Fittings 150
LÖF, G. O. G., Discussion on Design and Performance of a Residential Earth Heat Pump 391
LOSCH, J., TOHRU INOUYE, F. K. HICK, M.D., R. W. KEETON, M.D., and NATHANIEL GLICKMAN, A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment 35

M

MAHER, R. L., Discussion on Further Studies of the Thermal Characteristics of Plaster Panels 522
MARSCHALL, P. J., Discussion on Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 344
MART, L. T., Discussion on Performance of a Forced-Draft Cooling Tower 325
Maximum Downward Travel of Heated Jets from Standard Long Radius *A.S.M.E.* Nozzles, by LINN HELANDER, SHEE-MANG YEN and R. E. CRANK 241
MCKINLEY, R. W., Discussion on Design Data for Slat-Type Sun Shades for Use in Load Estimating 433
MCNALL, P. E., JR., Discussion on Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F 345
MCRAE, M. W., Discussion on Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems 359
Mechanical Condensing Unit, Cooling a Small Residence with a Two-Horsepower, by H. T. GILKEY, D. R. BAHNFLETH and R. W. ROOSE 283

MEEK, G. W., Discussion on The Bacteria and Odor Control Problem in Occupied Spaces 90
Metal Duct and Fittings, Pressure Losses in 4-inch Diameter Galvanized, by H. G. CONN, W. G. COLBORNE and W. G. BROWN 139
MILLS, D. L., Discussions on: Further Studies of the Thermal Characteristics of Plaster Panels 521
Heat Exchanges in a Ceiling Panel Heated Room 216
Thermal Performance of Frame Walls 460
MINDAK, R. J., Discussion on Heat Exchanges in a Ceiling Panel Heated Room 218
Moisture Movement in Soils Due to Temperature Difference, by W. A. HADLEY and RAYMOND EISENSTADT 395
Discussion 401
MURPHY, H. C., Discussion on The Bacteria and Odor Control Problem in Occupied Spaces 88
MYLER, W. M., JR., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 490

N

NAMAN, I. A., Discussion on Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit 304
NEWTON, C. G., Discussion on Heat Exchanges in a Floor Panel Heated Room 509
Noise and Noise Transmission in Piping Systems, Experimental Approaches to the Study of, by W. L. ROGERS 347
NOTTAGE, H. B., C. V. FRANKS, L. E. HULBERT and L. F. SCHUTRUM, Heat Flow Analysis in Panel Heating or Cooling Sections 527
NOTTAGE, H. B., Discussions on: Air Velocities in Ventilating Jets 282
The Bacteria and Odor Control Problem in Occupied Spaces 91
Heat Exchanges in a Ceiling Panel Heated Room 217
Nozzles, Standard Long Radius *A.S.M.E.*, Maximum Downward Travel of Heated Jets from, by LINN HELANDER, SHEE-MANG YEN and R. E. CRANK 241

O

Odor Control Problem in Occupied Spaces, The Bacteria and, by R. L. KUEHNER 77
OSLUND, D. L., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 491

P

Panel Heated Room, Ceiling, Heat Exchanges in a, by L. F. SCHUTRUM, G. V. PARMELEE and C. M. HUMPHREYS 197
Panel Heated Room, Floor, Heat Exchanges in a, by L. F. SCHUTRUM, G. V. PARMELEE and C. M. HUMPHREYS 495
Panel Heating or Cooling Sections, Heat Flow Analysis in, by H. B. NOTTAGE, C. V. FRANKS, L. E. HULBERT and L. F. SCHUTRUM 527
PARMELEE, G. V., and D. J. VILD, Design Data for Slat-Type Sun Shades for Use in Load Estimating 403
PARMELEE, G. V., Discussion on Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit 303
PARMELEE, G. V., L. F. SCHUTRUM and C. M. HUMPHREYS, Heat Exchanges in a Ceiling Panel Heated Room 197
PARMELEE, G. V., L. F. SCHUTRUM and C. M. HUMPHREYS, Heat Exchanges in a Floor Panel Heated Room 495
Performance of a Forced-Draft Cooling Tower, by B. H. SPURLOCK, JR. 311
Discussion 325
Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence, by H. T. GILKEY, R. W. ROOSE and M. E. CHILDS 473
Discussion 489
Perimeter-Loop and Perimeter-Radial Systems in a Residence, Performance of Warm-Air, by H. T. GILKEY, R. W. ROOSE and M. E. CHILDS 473

Permeance Measurement by the Permeometer, Automatic, by F. A. JOY and A. W. SHERDON 435
PHIFER, J. E., and RUTCHER SKAGERBERG, Fuel Consumption Analysis for Multi-Family Housing Projects 113
Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment, A Comparison of, by TOHRU INOUYE, F. K. HICK, M.D., R. W. KEETON, M.D., J. LOSCH and NATHANIEL GLICKMAN 35
Pipe Lines, Two Methods of Calculating the Friction in, by F. E. GIESECKE 49
Piping Systems, Experimental Approaches to the Study of Noise and Noise Transmission in, by W. L. ROGERS 347
Plaster Panels, Further Studies of the Thermal Characteristics of, by L. F. SCHUTRUM and C. M. HUMPHREYS 511
Pressure Losses in 4-inch Diameter Galvanized Metal Duct and Fittings, by H. G. CONN, W. G. COLBORNE and W. G. BROWN 139
Discussion 150
Proceedings, Annual Meeting 1
Semi-Annual Meeting 305
Program, Fifty-Ninth Annual Meeting 31
Semi-Annual Meeting, 1953 308

R

REED, M. J., Discussion on Fuel Consumption Analysis for Multi-Family Housing Projects 125
Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F, by TOHRU INOUYE, F. K. HICK, M.D., S. E. TELSER, M.D., and R. W. KEETON, M.D. 329
Research, Committee on, Annual Report of 18
Residence With a Two-Horsepower Mechanical Condensing Unit, Cooling a Small, by H. T. GILKEY, D. R. BAHNFLETH and R. W. ROOSE 283
Residential Earth Heat Pump, Design and Performance of a, by MERL BAKER 371
ROGERS, W. L., Experimental Approaches to The Study of Noise and Noise Transmission in Piping Systems 347
Room Air Distribution Research for Year 'Round Air Conditioning, Part I, by S. F. GILMAN, H. E. STRAUB, A. E. HERSHEY and R. B. ENGDAHL 151
ROOSE, R. W., H. T. GILKEY and D. R. BAHNFLETH, Cooling a Small Residence With a Two-Horsepower Mechanical Condensing Unit 283
ROOSE, R. W., H. T. GILKEY and M. E. CHILDS, Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence 473
ROSS, SHERMAN, Discussion on The Bacteria and Odor Control Problem in Occupied Spaces 92

S

SANFORD, S. S., Discussion on Design and Performance of a Residential Earth Heat Pump 392
SARTAIN, E. L., Discussion on Further Studies of the Thermal Characteristics of Plaster Panels 522
SCHUTRUM, L. F., and C. M. HUMPHREYS, Further Studies of the Thermal Characteristics of Plaster Panels 511
SCHUTRUM, L. F., G. V. PARMELEE and C. M. HUMPHREYS, Heat Exchanges in a Ceiling Panel Heated Room 197
SCHUTRUM, L. F., G. V. PARMELEE and C. M. HUMPHREYS, Heat Exchanges in a Floor Panel Heated Room 495
SCHUTRUM, L. F., H. B. NOTTAGE, C. V. FRANKS and L. E. HULBERT, Heat Flow Analysis in Panel Heating or Cooling Sections 527
Semi-Annual Meeting, 1953 305
Committee on Arrangements 310
Program 308
Resolutions on 307

Shades, Sun, Design Data for Slat-Type, for Use in Load Estimating, by G. V. PARMELEE and D. J. VILD ... 403
SHERDON, A. W., and F. A. JOY, Automatic Permeance Measurement by the Permeometer ... 435
SIU, R. G. H., Discussion on The Bacteria and Odor Control Problem in Occupied Spaces ... 89
SKAGERBERG, RUTCHER, and J. E. PHIFER, Fuel Consumption Analysis for Multi-Family Housing Projects ... 113
SKINNER, N. D., Discussion on Compression Tank Selection for Hot Water Heating Systems ... 74
Slat-Type Sun Shades for Use in Load Estimating, Design Data for, by G. V. PARMELEE and D. J. VILD ... 403
SNYDER, E. F., JR., A. B. ALGREN and J. S. LOCKE, Field Studies of Floor Panel Control Systems ... 173
Soils, Moisture Movement in, Due to Temperature Difference, by W. A. HADLEY and RAYMOND EISENSTADT ... 395
SPURLOCK, B. H., JR., Performance of a Forced-Draft Cooling Tower ... 311
STRAUB, H. E., S. F. GILMAN, A. E. HERSHEY and R. B. ENGDAHL, Room Air Distribution Research for Year 'Round Air Conditioning ... 151
SULLIVAN, R. R., Discussion on Automatic Permeance Measurement by the Permeometer ... 447
Symposium on Administration of Engineering Research ... 17

T

TASKER, CYRIL, Discussion on Air Velocities in Ventilating Jets ... 282
TASKER, CYRIL, Report of Director of Research ... 21
TASKER, CYRIL, Resolution on Death of ... 306
TELSER, S. E., M.D., TOHRU INOUYE, F. K. HICK, M.D., and R. W. KEETON, M.D., Effect of Relative Humidity on Heat Loss of Men Exposed to Environments of 80, 76, and 72 F ... 329
Temperature Difference, Moisture Movement in Soils Due to, by W. A. HADLEY and RAYMOND EISENSTADT ... 395
Thermal Characteristics of Plaster Panels, Further Studies of the, by L. F. SCHUTRUM and C. M. HUMPHREYS ... 511
Thermal Conductivity Probe, Development of the, by F. C. HOOPER and S. C. CHANG ... 463
Thermal Performance of Frame Walls, Part II—Air Spaces Blocked at Mid-Height, by G. O. HANDEGORD and N. B. HUTCHEON ... 449
Discussion ... 460
THULMAN, R. K., Discussions on: Heat Transmitted to the I=B=R Research Home from the Inside Chimney ... 111
Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence ... 491
Thermal Performance of Frame Walls ... 460
TOELAER, C. M., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence ... 489
TOULOUKIAN, Y. S., Author's Closure on Design and Performance of a Residential Earth Heat Pump ... 393
TOULOUKIAN, Y. S., Discussion on Moisture Movement in Soils Due to Temperature Difference ... 402
TURK, AMOS, Discussion on The Bacteria and Odor Control Problem in Occupied Spaces ... 91
TUVE, G. L., Air Velocities in Ventilating Jets ... 261
TUVE, G. L., Discussion on Thermal Performance of Frame Walls ... 460
Two Methods of Calculating the Friction in Pipe Lines, by F. E. GIESECKE ... 49

V

Ventilating Jets, Air Velocities in, by G. L. TUVE ... 261
VILD, D. J., and G. V. PARMELEE, Design Data for Slat-Type Sun Shades for Use in Load Estimating ... 403

W

WALKER, E. A., Participation by the Colleges in Engineering Research..... 17
WALLACE, W. M., II, Discussion on Further Studies of the Thermal Characteristics of Plaster Panels..... 522
Walls, Frame, Thermal Performance of, by G. O. HANDEGORD and N. B. HUTCHEON 449
Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence, Performance of, by H. T. GILKEY, R. W. ROOSE and M. E. CHILDS..... 473
WEBER, E. L., Discussions on: Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems..... 358
Further Studies of the Thermal Characteristics of Plaster Panels..... 523

Y

Year 'Round Air Conditioning, Room Air Distribution Research for, Part I, by S. F. GILMAN, H. E. STRAUB, A. E. HERSHEY and R. B. ENGDAHL..... 151
YEN, SHEE-MANG, LINN HELANDER and R. E. CRANK, Maximum Downward Travel of Heated Jets from Standard Long Radius *A.S.M.E.* Nozzles..... 241
YOUNG, P. W., Discussion on Performance of Warm-Air Perimeter-Loop and Perimeter-Radial Systems in a Residence..... 492

Z

ZEIL, H. E., Discussion on Performance of a Forced-Draft Cooling Tower..... 327

17
522
449
473
358
523
151
241
492
327